MATHEMATICAL MODELLING OF ROOF SURFACE EVAPORATION for LOW ENERGY COOLING

By

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CERTIFICATE

It is certified that the work contained in the thesis entitled **Mathematical Modelling of Roof Surface Evaporation for Low Energy Cooling,** by **Manoj Kumar Verma**, has been carried out under my supervision and that this work has not been submitted elsewhere for a degree.

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July, 1997

Dedicated to -

My Beloved Sister

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Synopsis

Name of Student: **Manoj Kumar Verma** Roll No. **9510521**
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Mathematical Modelling of Roof Surface Evaporation for Low Energy Cooling

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Roof surface evaporation and its effect on cooling load has been studied to provide low energy cooling in the residential, official and commercial buildings. A generalised mathematical model has been developed to calculate the hourly cooling load for buildings. The heat capacity of walls is taken into account by finite difference approach.

Results have been obtained for the case of roof surface evaporation. It shows that roof surface evaporation renders about 21 % saving in peak cooling load. Thermal stresses in building materials are reduced due to reduced thermal potential across the roof surfaces. Because evaporation over the roof surface renders the temperature of the roof to approach the wet-bulb temperature, which is almost constant and lesser than the dry-bulb temperature.

Results have been obtained for some typical inside conditions. The inside dry-bulb temperature has been maintained to be higher than the generally used value

for comfort air conditioning. However, the comfort level has been maintained by increasing the air velocity. By this technique the cooling load has been reduced without significantly sacrificing human comfort.

The mathematical model can help practice engineers select the appropriate capacity of air conditioning equipment for buildings with or without roof surface evaporation. The same is quite relevant in the time of inadequate availability of electricity as well as cost effective operation, being an important consideration these days in the competitive environment. The saving in electricity in the present work does not claim to provide a panacea, but it can contribute significantly towards, " Electricity saving is electricity generation ".

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Nomenclature

A	area of walls/roof, [m^2]
A_{door}	area of door, [m^2]
A_{sun}	glass area directly exposed to sun, [m^2]
A_{win}	total glass area of window, [m^2]
c	specific heat capacity of walls/roof, [kJ/kg-K]
CLF	cooling load factor
d	declination angle
F_{sg}	angle factor between surface and ground
F_{ss}	angle factor between surface and sky
h	hour angle
h_o	outside film coefficient, [W/m^2]
h_i	inside film coefficient, [W/m^2]
H	height of window, [m]

I_d	diffused solar radiation, [W/m^2]
I_{dg}	diffused solar radiation reflected from ground, [W/m^2]
I_{ds}	diffused solar radiation reflected from clear sky, [W/m^2]
I_D	direct solar radiation, [W/m^2]
I_{DN}	direct normal radiation, [W/m^2]
I_t	total solar radiation, [W/m^2]
I_{tH}	total solar radiation falling on ground, [W/m^2]
I_r	reflected solar radiation, [W/m^2]
k	thermal conductivity of walls/roof, [W/m-K]
k_g	thermal conductivity of window glass, [W/m-K]
l	latitude of the place
L	ifiltration/ventilation air in litre per second
no	number of people in the conditioned space
N_{ach}	number of air changes per hour
$\dot{Q}_{al}$	latent load due to appliances, [W]
$\dot{Q}_{as}$	sensible load due to appliances, [W]
$\dot{Q}_{door}$	heat transfer through door, [W]
$\dot{Q}_{glass}$	heat transfer through glass, [W]

$\dot{Q}_{il}$	latent load due to infiltration, [W]
$\dot{Q}_{is}$	sensible load due to infiltration, [W]
$\dot{Q}_{latent}$	total latent load, [W]
$\dot{Q}_{ls}$	sensible load due to lighting, [W]
$\dot{Q}_{ol}$	latent load due to occupancy, [W]
$\dot{Q}_{os}$	sensible load due to occupancy, [W]
$\dot{Q}_{ps}$	sensible load due to power equipments, [W]
$\dot{Q}_{sensible}$	total sensible load, [W]
$\dot{Q}_{ss}$	sensible solar load, [W]
$\dot{Q}_{structural}$	heat transfer through walls/roof, [W]
$\dot{Q}_{total}$	total cooling load, [W]
$\dot{Q}_{vl}$	latent load due to ventilation, [W]
$\dot{Q}_{vs}$	sensible load due to ventilation, [W]
$(\dot{Q}_{roof})_{with\ evap.}$	heat transfer through roof with surface evaporation, [W]
$(\dot{Q}_{total})_{with\ evap}$	total cooling load with surface evaporation, [W]
RH_i	inside relative humidity, [%]
S_H	shadow height due to horizontal projection, [m]
S_w	shadow width due to vertical projection, [m]

t_g	temperature of glass sheet, [^oC]
t_{gi}	temperature of inner surface of glass sheet, [^oC]
t_{go}	temperature of outer surface of glass sheet, [^oC]
t_i	inside air temperature, [^oC]
t_o	outside air temperature, [^oC]
t_{si}	inside surface temperature, [^oC]
t_{so}	outside surface temperature, [^oC]
T_{db}	dry-bulb temperature of outside air, [^oC]
T_{max}	maximum air temperature of a day, [^oC]
T_{min}	minimum air temperature of a day, [^oC]
T_{wb}	wet-bulb temperature of outside air, [^oC]
T_{sol}	sol-air temperature, [^oC]
U	overall film coefficient, [W/m^2]
V_i	inside air velocity, [m/s]
V_o	outside air velocity, [m/s]
V_{room}	volume of room, [m^3]
W	width of window, [m]
$WB_{depression}$	wet-bulb depression, [^oC]

β	solar altitude angle
θ	solar incidence angle
ϕ	solar azimuth angle
ψ	surface azimuth angle
γ	surface-solar azimuth angle
Σ	surface tilt angle from horizontal
Ω	profile angle
ϵ	emissivity of the surface
η_{se}	saturation efficiency
α	thermal diffusivity of walls/roof
α_d	absorptivity of glass for diffused radiation
α_D	absorptivity of glass for direct radiation
α_s	absorptivity of surface
τ_d	transmissivity of glass for diffused radiation
τ_D	transmissivity of glass for direct radiation
ρ	density of walls/roof, [kg/m^3]
ϱ_g	reflectance of foreground
ΔT	difference between inside and outside temperatures, [^oC]

Δx thickness of walls/roof, [m]

$\Delta \tau$ time interval, [s]

$\Delta \omega$ difference between inside and outside specific humidities

Chapter 1

Introduction

1.1 Description

In today's circumstances, Air Conditioning is no more luxury, even in India. Its demand is ever increasing due to excessive research in the sophisticated fields, computerization of various activities, more and more advanced manufacturing processes in this era of economic liberalization. But, the energy sector in India has its own limitations to meet this ever increasing demand for electricity from every corner of the country. Need of the day is to save as much electricity as possible from here and there keeping faith in this old Indian proverb, " BOOND BOOND SE BHARE TALAB ". In this light, this work intends to contribute its part.

If a compromise is made in the comfort level, just a little bit and the higher dry-bulb temperature with enhanced air velocity are selected as indoor conditions, significant reduction in cooling load requirement is seen. This has been supported by researchers [1,2]. This concept of using high air velocity known as "Spot Cooling" is utilized in industries, air-crafts, offices, etc. in order to achieve both comfort and energy saving.

In India, Malhotra [3] has obtained the effective temperatures for hot & humid as well as hot & dry climate based on recording of votes by subjects about their

sensation in different working environments. Table 1.1 illustrates the comfort zone found by him.

Table 1.1: Recommended Effective Temperatures.

Sl. No.	Level of Comfort	Effective Temperature	
		Hot humid climate	Hot dry climate
1.	Warm and unpleasant	27.0	26.0 - 28.3
2.	Comfortable and pleasant (upper level)	24.0 - 25.0	24.4 - 26.6
3.	Comfortable and pleasant (lower level)	22.0 - 22.5	21.1 - 24.3

A minimum outdoor air supply rate of 2.5 L/s (5 cfm) per person was recommended by Wood [4] and this is current standard [1] also [5]. This reduces the cooling load requirement.

Usually, the cooling load calculation is based on only design outside maximum and minimum temperatures. Several researchers have considered daily hourly temperature by sinusoidal function. With a view to make it more realistic approach, the ASHRAE Handbook of Fundamentals [5] suggests an hourly temperature as the percentage of daily range for the cooling load calculation. It means one should not necessarily have a complete temperature time history of the place. Only maximum and minimum temperature of the day is required to calculate the cooling load. In the current work, ASHRAE procedure has been used.

[1]The latest tests has revealed that though this reduction in the standard ventilation requirement is theoretically sufficient, the field results from the U. S. army termed it as unsatisfactory level of air quality from health consideration in the long run.

1.2 Literature Review

Exhaustive studies have been done in the field of air conditioning systems, evaporative cooling and surface evaporation process and few of them are discussed in the subsequent paragraphs :

1.2.1 Analysis of Cooling Load Calculation

The analysis of cooling load involves parameters like geographical location, orientation and structural characteristics of building, ambient air temperature, relative humidity, etc.

The following three approaches have been developed to calculate the structural load of a building :

- Threlkeld's Classical approach [6]
- Transfer Function approach or Cooling Load Temperature Difference (CLTD) approach [7]
- Finite Difference approach [8]

Threlkeld's Classical approach [6] is based on the concept of sol-air temperature. To consider the effect of the thermal capacity of wall/roof on the heat transfer, two parameters (1) time lag (2) decrement factor have been devised. The values of these parameters (time lag and decrement factor) vs. thickness of wall/roof have been plotted for the standard wall/roof construction and their use for load calculation is employed.

In the Transfer Function approach [7], the various components of space heat gain are added together to get an instantaneous total rate of space heat gain. It is then converted into cooling load through the use of weighting factors called "*room transfer functions*". The transfer function is nothing more than a set of coefficient that relate to an output function at a given time to the value

of one or more driving functions. First, the cooling load is calculated for the standard walls/roof by Transfer Function approach, then standard equivalent CLTDs are obtained by dividing the cooling load by the U-factor for each wall or roof. For a given wall/roof (similar in thermal mass), the cooling load can be obtained by multiplying the total CLTDs by the U-factor. The error introduced by this approach depend on the extent of the difference between the construction in question and the one used for calculating the temperature differences. Several correction factors are employed to minimize the error.

Finite Difference approach has been described by Kadambi and Hutchinson [8]. They have approximated the structural load calculation by considering one dimensional transient heat transfer through walls and roof. This numerical approach has got impetus due to availability of digital computers. The basic simplicity of this approach compared with analytical solution is asserted in their work.

In view of tremendous computer facility at IIT K, the present work has been carried out using Finite Difference approach in order to predict the accurate cooling load for any residential building. The detailed procedure of cooling load calculation, from the point of view of practice engineers is outlined in ASHRAE Handbook of Fundamentals [5].

1.2.2 Surface Evaporation

In the light of energy crisis in India, the need of the day is to use an economic methods of evaporative cooling.

Leonardo da Vinci built a water powered evaporative cooler for the bed room of his patron's wife, first time. It gave reasonable comfort inside the room.

The mechanical direct evaporative cooler was developed in about 1932. Recently Jain [9] did extensive work in the field of surface evaporation at CBRI, Roorkee. He employed either gunny bag or organic fibrous material on roof terraces to keep a wet layer over the roof. An electronic sensor was developed

for automatic starting of the pump for wetting of the organic fibrous material. The same approach is adopted in the present work.

The characteristics of dry tropical climate are given in Jannot Yves [10]. The needs of economic equipment for air cooling are outlined from climatic data and local financial feasibilities. For northern part of India, an acceptable thermal comfort can be maintained by direct evaporative cooling of the outside air.

1.3 Present Work

A generalized computer programme has been developed for the accurate prediction of the hourly cooling load for a residential building located in any part of the northern hemisphere. The calculations are carried out using actual as well as approximated daily hourly temperatures. The approximated daily hourly temperature variation is predicted from the known maximum and minimum outdoor temperature [11] with the help of procedure given in the ASHRAE Handbook of Fundamentals [5].

After the evaluation of structural load of the building, the computer programme incorporates for the determination of overall cooling load. This would enable the air conditioning practice engineers to select the appropriate capacity of air conditioning unit.

The indoor design condition has been kept higher than the normally prescribed values without violeting the human comfort requirement. Moreover, due to more concern for energy conservation, the roof surface evaporation has been envisaged in same programme and a saving in the cooling load has been found to the tune of 21 %. In addition to cheap cooling technique, the roof surface evaporation reduces the thermal stress in the building.

Chapter 2

Cooling Load Calculation

2.1 Different Heat Sources :

The cooling load for air conditioning systems for summer, in general, comprises:

- *Sensible Heat Load*
- *Latent Heat Load*

The cooling load arises from the following sources:

1. External Sources :

   (a) Heat transmission through barriers such as walls, doors, ceiling, floor, etc. being caused by the temperature difference existing on the two sides of the barrier.

   (b) The solar heat, absorbed by walls, roofs and windows exposed to radiation from the sun and transferred to the inside space.

   (c) Heat and moisture introduced to the conditioned space through infiltration and ventilation air.

2. Internal Sources :

    (a) Occupancy load, both sensible and latent

    (b) Lighting load

    (c) Power equipment load, e.g., fans, water pumps, etc.

    (d) Appliances load, both sensible and latent

2.2 Need for Hourly Cooling Load Calculations

It is a well known fact that the ambient air temperature is not constant in a day, and accordingly the cooling load varies over a period of 24 hours. This is mainly caused by the variation in the solar intensity falling on the earth surface. The load calculation is further complicated by the fact that a wall has thermal capacity, due to which a certain amount of heat passing through it is stored and is transmitted to the inside at some time latter. Therefore, calculation based on instantaneous heat transmission through structure without considering the thermal capacity of the wall, is erroneous. Moreover, the traditional methods of evaluating the cooling load involves various assumptions like the load from each component is constant in a day and maximum value of the cooling load is equal to the individual maximum, which, in general, is not correct, as the occurrence of maximum load at particular time is not the occurrences of the other components. It is proper to adopt an hourly cooling load calculation to facilitate an economic selection of an air conditioning unit.

2.3 Structural Load :

2.3.1 Building Survey

The building survey incorporates the flowing particulars:

1. Location of the building ,i.e., longitude and latitude
2. Orientation
3. Dimension of the building structure, such as length, breadth , height and thickness of each layer of the building material
4. Composition of building material and their physical properties.

2.3.2 Hourly Outside Temperature Variation

Figures 2.1 to 2.4 show the hourly variation in outside temperature for Kanpur. The meteorological data reveals that the minimum temperature occurs just one or two hours before sunshine while the maximum temperature occurs three to four hours after the solar noon [11]. Since, the temperature time history is available for only a few places, an hourly variation in temperature is predicted based on maximum and minimum temperatures.

The ASHRAE Handbook of Fundamentals [5] has outlined a procedure to predict the hourly temperature. In the present work, the same is followed.

$$t_o = T_{max} - DailyRange \times PercentageFactor/100 \tag{2.1}$$

where, t_o=temperature at any time in ^oC

$DailyRange = T_{max} - T_{min}$

The percentage factor is taken for that hour for which outside temperature is to be calculated. These factors are tabulated in [5] and are reproduced here in Appendix 'A'.

The comparison of the temperature variation by theoretical with actual variation is shown in figures 2.1 to 2.4. The data have been taken from the average temperature of three years, i.e., 1982, 1983 and 1985 [11], and are reproduced here in Appendix 'B'.

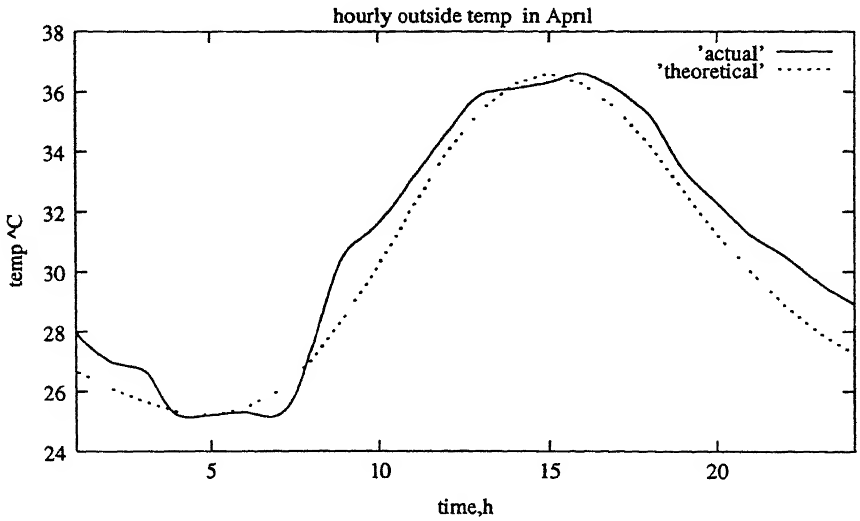

Figure 2.1: Hourly variation of actual and theoretical temperature in April.

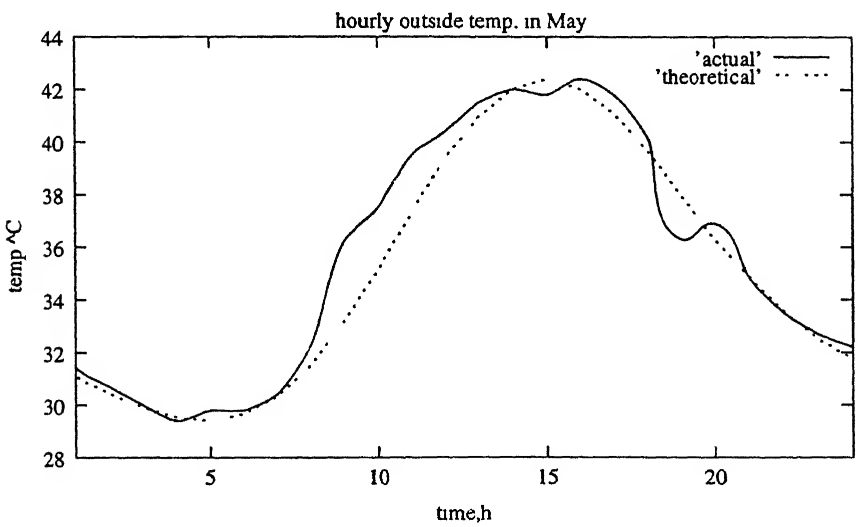

Figure 2.2: Hourly variation of actual and theoretical temperature in May.

2.3.3 Solar Radiation

Solar radiation forms the significant part of cooling load for the residential buildings. The total radiation (I_t), reaching on terrestrial surface is the sum of the direct solar radiation (I_D), the diffused sky radiation (I_d) and the solar radiation reflected from the surrounding surface (I_r). The intensity of the direct component is the product of the direct normal radiation (I_{DN}) and the

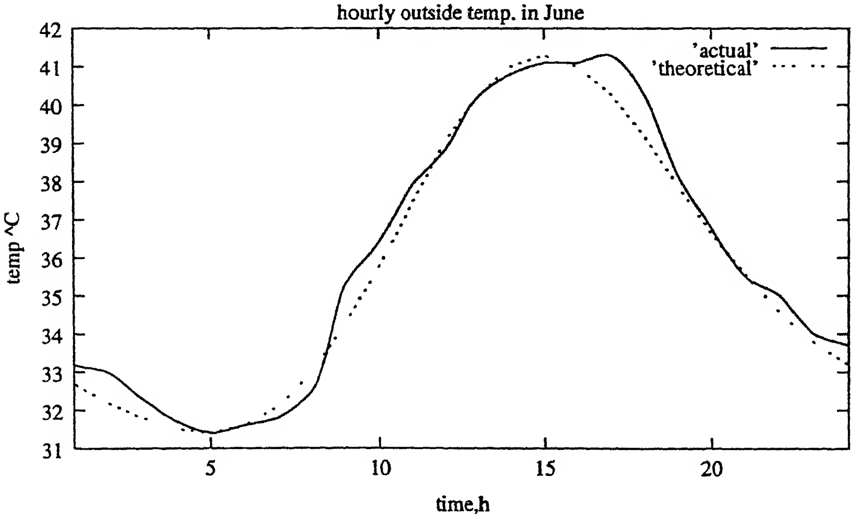

Figure 2.3: Hourly variation of actual and theoretical temperature in June.

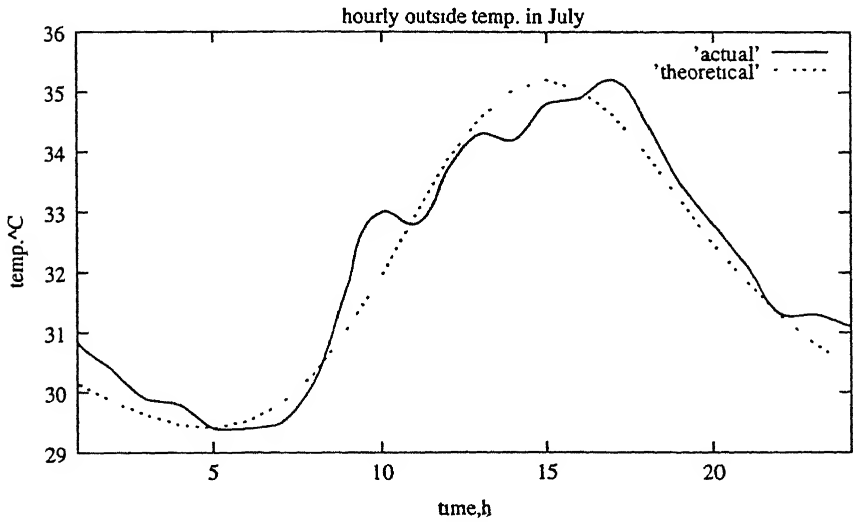

Figure 2.4: Hourly variation of actual and theoretical temperature in July.

cosine of the incidence angle (θ) between the incoming solar rays and a line normal to the surface as shown in figure 2.5. Thus,

$$I_t = I_D + I_d + I_r \tag{2.2}$$

In the present analysis the reflected radiation (I_r) is neglected.

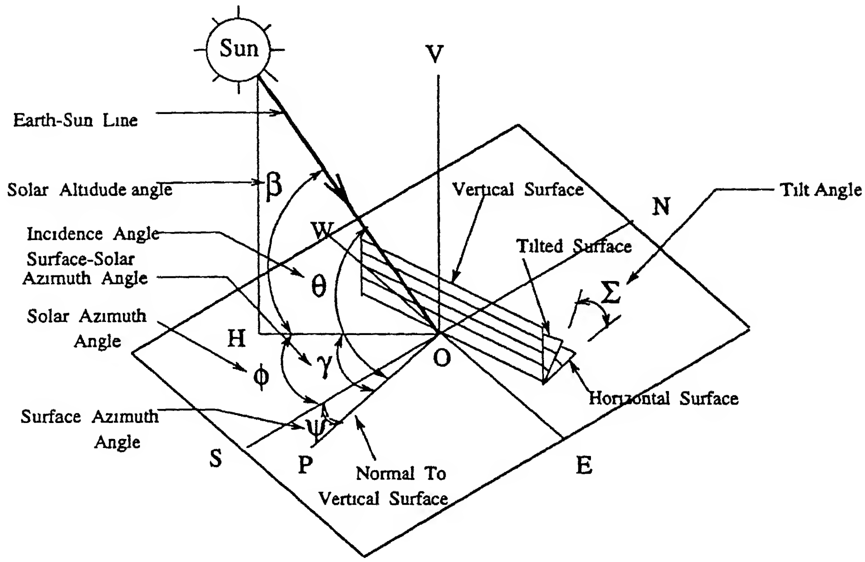

Figure 2.5: Solar angles for vertical and horizontal surfaces.

I_D is given as:

$$I_D = I_{DN} \times cos\theta \tag{2.3}$$

where, I_{DN} is expressed as :

$$I_{DN} = \frac{A}{exp(B/sin\beta)} \tag{2.4}$$

I_d consists two parts:

1. diffused solar radiation from clear sky

$$I_{ds} = C.I_{DN}.F_{ss} \tag{2.5}$$

where, F_{ss} is the angle factor between the surface and the sky, i.e., the fraction of short wave radiation emitted by the sky that reaches the tilted surface (dimensionless). F_{ss} is 0.5 for vertical surfaces and 1.0 for horizontal surfaces. For other surfaces,

$$F_{ss} = (1 + \cos\Sigma)/2 \tag{2.6}$$

where, Σ is the tilt angle from thc horizontal plane(Fig 2.5).

2. ground reflected diffused solar radiation

$$I_{dg} = I_{tH}.\varrho_g.F_{sg} \tag{2.7}$$

where, ϱ_g is the reflectance of foreground and F_{sg} is the angle factor between the surface and the ground (dimensionless), and $F_{sg} = 1 - F_{ss}$. I_{tH} represents the total solar radiation falling on the ground, and is given as$I_{tH} = I_{DN}(C + \sin\beta)$.

The value of A, B and C are taken from [5] and the same are reproduced in Appendix 'C' for the twenty first day of the each month.

2.3.4 Determination of Angles of Incidence

The sun's position in sky is most conveniently expressed in terms of solar altitude β, above the horizontal and solar azimuth angle ϕ, measured from south. These angles β and ϕ are expressed in the terms of the latitude of the place l, solar declination d, and hour angle h.

$$sin\beta = cosl.cosd.cosh + sinl.sind \tag{2.8}$$

and,

$$cos\phi = \frac{sin\beta.sinl - sind}{cos\beta.cosl} \tag{2.9}$$

The hour angle (in degree) is calculated as :

$$h = (12 - LST) \times 15 \tag{2.10}$$

If a surface is tilted by an angle Σ to the horizontal plane, then the incidence angle θ is given by

$$cos\theta = cos\beta.cos\gamma.sin\Sigma + sin\beta.cos\Sigma \tag{2.11}$$

When the surface is horizontal, $\Sigma = 0^o$, and :

$$cos\theta_H = sin\beta \tag{2.12}$$

For a vertical surface, $\Sigma = 90^o$, and :

$$cos\theta_V = cos\beta.cos\gamma \tag{2.13}$$

The surface solar azimuth angle γ is given as :

$$\gamma = \phi - \psi \quad \text{, before solar noon} \tag{2.14}$$

$$\gamma = \phi + \psi \quad \text{, after solar noon} \tag{2.15}$$

where, ψ is surface azimuth angle which represents the orientation of the surface. The ψ values for different orientation are $\psi_S(0^o)$, $\psi_E(90^o)$, $\psi_N(180^o)$ and $\psi_W(270^o)$.

2.3.5 Problem Formulation

2.3.5.1 Heat Transfer through Walls and Roof

Heat Transmission through the wall or roof of building structures is not steady and is therefore, difficult to evaluate. The two principal factors causing this are:

1. The variation in the outside air temperature over a period of 24 hours.
2. The variation in the solar radiation intensity that is incident upon the surface over a period of 24 hours.

The phenomenon is further complicated by the fact that a wall or roof has a thermal capacity due to which a certain amount of heat passing through it is stored and is transmitted to outside and/or inside at some later time.

The problem requires a solution of the governing equation for unsteady-state one-dimensional heat transfer, viz.,

$$\frac{\partial t}{\partial \tau} = \alpha \times \frac{\partial^2 t}{\partial x^2} \tag{2.16}$$

where, t is the temperature at any section of the wall or roof at a distance x from the surface at a time τ, and α is the thermal diffusivity given by

$$\alpha = \frac{k}{\rho.c} \tag{2.17}$$

where, k is the thermal conductivity and $\rho.c$ is the heat capacity of the wall or roof, in which ρ and c are the density and specific heat, respectively.

The heat transfer equation is to be solved with the boundary conditions of periodic variation of outside air temperature and solar radiation.

2.3.6 Sol-air Temperature

The calculation of structural load as a result of heat gain through exterior roof and walls involves the concept of sol-air temperature [6]. A heat balance at a sunlit surface gives the heat flux to the surface as:

$$q = \alpha_s I_t + h_o(T_o - T_{so}) - \epsilon\Delta R \tag{2 18}$$

ϵ = emissivity of the surface, 1 0 for black body

ΔR=63.0 W/m^2, for horizontal surface and 0, for vertical surface [5]

α_s = absorptivity of the surface Equation 2.17 can also be written as

$$q = h_o(T_{sol} - T_{so}) \tag{2.19}$$

then, from equations 2 17 and 2.18 :

$T_{sol} = T_o + \alpha_s\ I_t/h_o$ - $\epsilon\ \Delta R/h_o$

Thus, the combined effect of the solar radiation and outside air temperature has been incorporated into a single effective temperature known as *sol-air temperature.*

2.3.7 Outside and Inside Film Coefficients for Walls and Roof

The outside film coefficients are given by [8,16]:

For exterior walls

$$h_o = 7\ 373 + 5.066 \times V_o \quad , (W/m^2.^oC) \tag{2.20}$$

For roof

$$h_o = 7\ 9953 + 6\ 364 \times V_o \quad , (W/m^2.^oC) \tag{2.21}$$

where, V_o is the outside air velocity in m/s.

The inside film coefficients are given by [8]:

For interior walls

$$h_i = 1.77 \times (\Delta t)^{0\ 25} + (7.373 + 5\ 066 \times V_i) \quad , (W/m^2.^oC) \tag{2.22}$$

For roof

$$h_i = 1.31 \times (\Delta t)^{0\ 25} + (7.9953 + 6.364 \times V_i) \quad , (W/m^2\ ^oC) \tag{2.23}$$

where,
$\Delta t = |t_{si} - t_i|$,
t_{si} = inside surface temperature,
t_i = inside temperature, and
V_i = inside air velocity in m/s.

The Newton Raphson iterative procedure has been used to get Δt and therefore, inside film coefficient (h_i).

2.3.8 Finite Difference Approach for Heat Transfer through Structure

The analytical solution for heat transfer through building structure is continuous but cumbersome and approximate due to rejection of a large number of terms to make it possible to find a mathematical solution.

For a numerical solution, the system is subdivided into discrete regions, each with a representative location (a single plane) in space. A reasonable time increment, which will be limited by the nature of the problem, must be chosen. For the sake of simplicity, in the present work, the system is subdivided into three different regions (e.g., outer cement plaster, brick work, and inner cement plaster) as usually find in the general constructions.

Figures 2.6 and 2.7 give the structural detail of walls and roof, respectively.

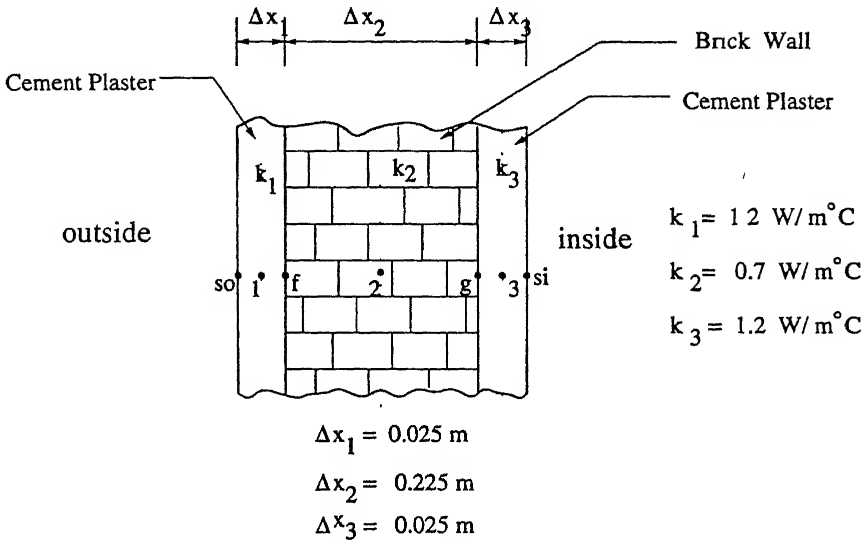

Figure 2.6: Structural detail of wall.

The salient features of the Finite Difference Approach is given bellow:

- $$\Delta\tau_{min} = \frac{(\Delta x_{min})^2}{4.0 \times \alpha_{max}} \tag{2.24}$$

  where, Δx_{min} is the minimum thickness of layers of walls/roof while α_{max} is the maximum thermal diffusivity of these layers.

  Select a convenient time interval($\Delta\tau$) which should be less than $\Delta\tau_{min}$.

  $$\Delta\tau_1 = \Delta\tau_2 = \Delta\tau_3 = \Delta\tau \tag{2.25}$$

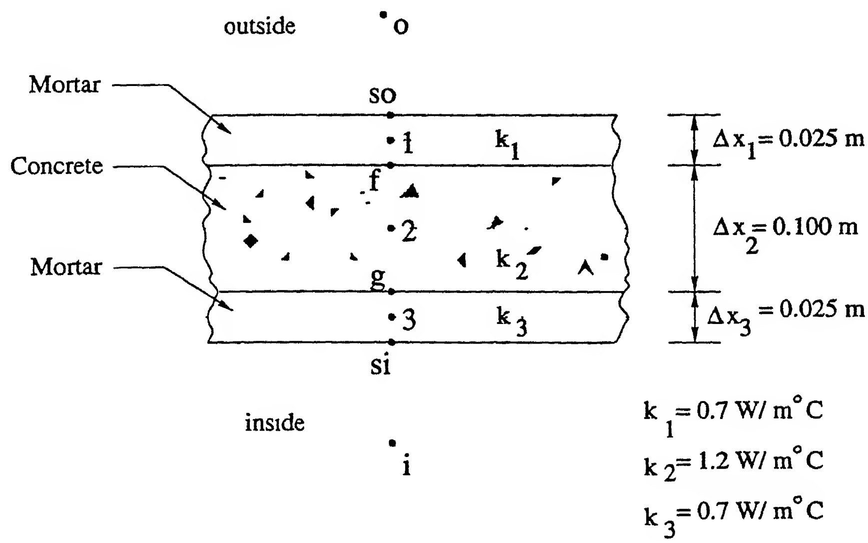

Figure 2.7: Structural detail of ceiling.

- Calculate following parameters.

$$m_1 = \frac{(\Delta x_1)^2}{\alpha_1 \times \Delta \tau_1} \tag{2.26}$$

$$m_2 = \frac{(\Delta x_2)^2}{\alpha_2 \times \Delta \tau_2} \tag{2.27}$$

$$m_3 = \frac{(\Delta x_3)^2}{\alpha_3 \times \Delta \tau_3} \tag{2.28}$$

$$Bi_o = \frac{h_o \times \Delta x_1}{k_1} \tag{2.29}$$

$$N_{12} = \frac{k_2 \times \Delta x_1}{k_1 \times \Delta x_2} \tag{2.30}$$

$$N_{23} = \frac{k_3 \times \Delta x_2}{k_2 \times \Delta x_3} \tag{2.31}$$

- Set initial condition

$$t_1(1) = ti \tag{2.32}$$

$$t_2(1) = ti \tag{2.33}$$

$$t_3(1) = ti \tag{2.34}$$

- Calculation of temperatures at every time interval $(\Delta\tau)$.

$$t_{so}(j) = \left(\frac{Bi_o}{Bi_o+2}\right).t_e(j) + \left(\frac{2}{Bi_o+2}\right).t_1(j) \tag{2.35}$$

$$t_f(j) = \left(\frac{1}{1+N_{12}}\right).t_1(j) + \left(\frac{N_{12}}{1+N_{12}}\right).t_2(j) \tag{2.36}$$

$$t_g(j) = \left(\frac{1}{1+N_{23}}\right).t_2(j) + \left(\frac{N_{23}}{1+N_{23}}\right).t_3(j) \tag{2.37}$$

Now, by Newton Raphson method, calculate inside film coefficient$(h_i(j))$ and inside surface temperature $(t_{si}(j))$.

Temperatures at time interval $(\tau + \Delta\tau)$

$$t_1' = (t_{so}(j) + t_f(j)).\frac{2}{m_1} + \left(1 - \frac{4}{m_1}\right).t_1(j) \tag{2.38}$$

$$t_2' = (t_f(j) + t_g(j)).\frac{2}{m_2} + \left(1 - \frac{4}{m_2}\right).t_2(j) \tag{2.39}$$

$$t_3' = (t_g(j) + t_{si}(j)).\frac{2}{m_3} + \left(1 - \frac{4}{m_3}\right).t_3(j) \tag{2.40}$$

Change initial values

$$t_1(j+1) = t_1' \tag{2.41}$$

$$t_2(j+1) = t_2' \tag{2.42}$$

$$t_3(j+1) = t_3' \tag{2.43}$$

Continue for the rest of the day.

- Repeat the temperature calculations, till the agreement occurs on the next day at $1A.M.$.

- Heat transfer through the walls or roof is given by

$$\dot{Q}_{structural} = h_i(j) \times A \times (t_{si}(j) - t_i) \tag{2.44}$$

2.3.9 Heat Transfer through Glass

Glass construction forms a significant part of modern building structures. The heat transfer through glass [12] comprises:

- all the transmitted radiation
- a part of the absorbed radiation that enters the conditioned space, and
- the heat transmitted due to temperature difference between the outside and inside temperatures.

The direct radiation enters the space only if the glass receives the direct rays of the sun. The diffuse radiation enters the space even when the glass is not facing the sun. Figure 2.8 represents all sorts of heat transfer into conditioned space through glass. The heat transfer to the space is give by :

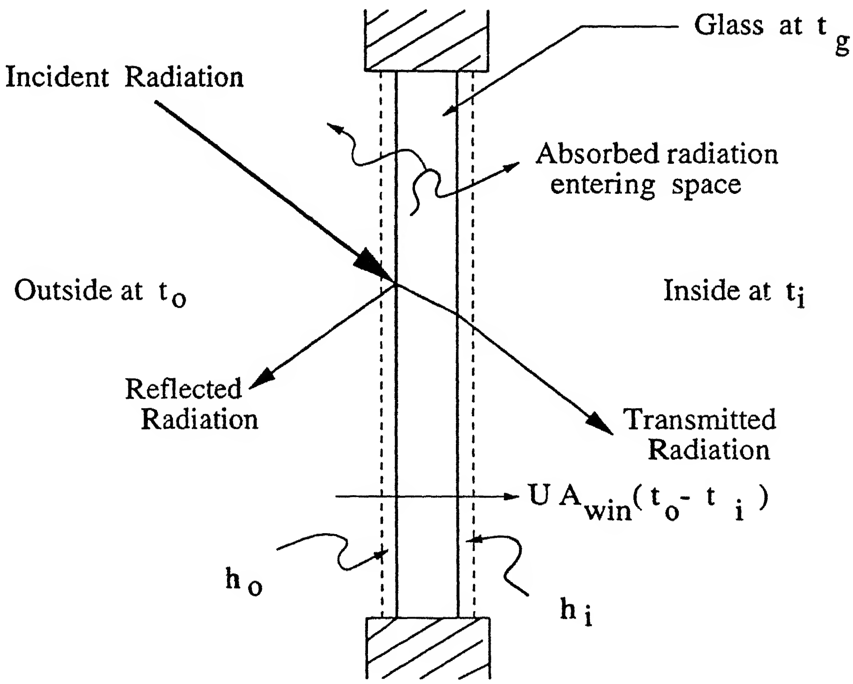

Figure 2.8: Heat transfer through glass.

$$\dot{Q}_{glass} = A_{sun}.\tau_D.I_D + A_{win}.\tau_d.I_d + h_i.A_{win}.(t_{gi} - t_i) \tag{2.45}$$

where,
t_{gi} = temperature of inner surface of glass
A_{sun} = glass area directly exposed to the sun
A_{win} = total glass area of window
The subscripts D and d denote the terms for direct and diffuse radiations, respectively.

Writing energy balance for glass sheet itself, we have

$$A_{sun}.\alpha_D.I_D + A_{win}.\alpha_d.I_d = A_{win}[hi.(t_{gi} - t_i) + h_o.(t_{go} - t_o)] \tag{2.46}$$

where, t_{go}=temperature of outer surface glass

As the thickness of glass sheet is very small, inner and outer surface temperatures of glass can be assumed to be equal to (t_g), without creating appreciable error in cooling load but simultaneously simplifying the analysis tremendously. Now, modified equation can be expressed as :

$$t_g = \frac{A_{sun}.\alpha_D.I_D + A_{win}.\alpha_d.I_d + A_{win}.h_i.t_i + A_{win}.h_o.t_o}{A_{win}(h_i + h_o)} \tag{2.47}$$

Putting the value of t_g in place of t_{gi} in equation (2.45) ,we get the cooling load due to glass as :

$$\dot{Q}_{glass} = A_{sun}.\tau_D.I_D + A_{win}.\tau_d.I_d + \frac{A_{sun}.\alpha_D.I_D + A_{win}.\alpha_d.I_d}{\left(1. + \frac{h_o}{h_i}\right)} + U.A_{win}.(t_o - t_i) \tag{2.48}$$

where, U is the overall coefficient of heat transfer given by

$$\frac{1}{U} = \frac{1}{h_o} + \frac{1}{h_i} \tag{2.49}$$

If the thermal resistance of glass is also considered, U is given by :

$$\frac{1}{U} = \frac{1}{h_o} + \frac{\Delta x}{k_g} + \frac{1}{h_i} \tag{2.50}$$

where, Δx is the thickness of glass and k_g is its thermal conductivity.

Here, it has to be kept in mind that the transmissivity and absorptivity are function of the angle of incidence. They vary for direct radiation while are almost constant for diffused part. These values are given in Appendix 'D'[13].

2.3.10 Shading of Surfaces

The most effective way to reduce the solar load on fenestration is to intercept direct radiation from the sun before it reaches the glass. To serve this purpose, most glass areas are provided with reveals, overhangs and fins in the form of vertical and horizontal projections from the walls.

The ability of these projections to intercept the direct component of solar radiation depend on their geometry, surface-solar angle (γ) and the profile or shadow-line angle (Ω) (Fig 2.9). The profile angle is defined as the angular difference between a horizontal plane and a plane tilted about a horizontal axis in the plane of the fenestration until it includes the sun. The profile angle can be calculated by:

$$tan\Omega = \frac{tan\beta}{cos\gamma} \tag{2.51}$$

The shadow height, S_H, due to horizontal projection, P, on a window or wall

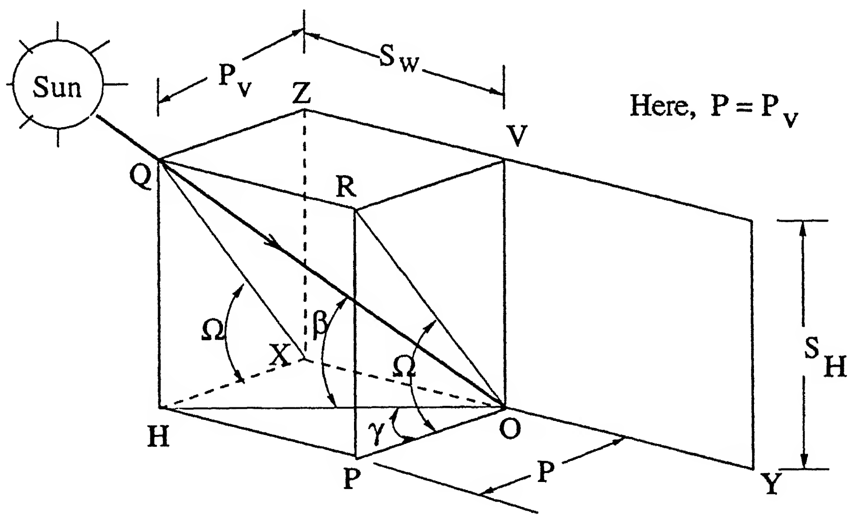

Figure 2.9: Shading on glass due to horizontal and vertical projections.

for any time of day or year is related to the profile angle Ω by:

$$S_H = P \times cot\Omega \tag{2.52}$$

The shadow width S_w, due to a vertical projection, P_v, (it need not not be equal to the horizontal projection) on a window or wall for any given time of

day and year is related to the vertical surface -solar azimuth angle γ by:

$$S_w = P_v \times cot\gamma \tag{2.53}$$

Thus, the sunlit area of the window is:

$$A_{sun} = (W - S_w).(H - S_H) \tag{2.54}$$

where, W and H are the width and height of window.

2.3.11 Heat Transfer through Door

The cooling load due to doors are given by

$$\dot{Q}_{door} = U.A_{door}.(t_o - t_i) \tag{2.55}$$

where, U is the overall heat transfer coefficient through door [5] and A_{door} is area of the door.

2.4 Infiltration Load

Infiltration is the uncontrolled flow of air through openings in the building envelop driven by pressure difference across the shell. Infiltration is balanced by an equal amount of exfiltration since, except for transient conditions, there is no net storage of air in a building.

There are several techniques to calculate the rate of infiltration :

Air Change Method : The air change method for estimating infiltration is based on past experience. It applies to average residential construction under average weather conditions.

ASHRAE Handbook of Fundamentals [5] presents the number of air changes per hour to be expected in rooms with varying exposures to

give the air leakage rates. The infiltration rates for average building under average conditions can be assumed to be one half of that values.

This method has been used in present work because of its simplicity.

The air exchange method has a size effect. A large room with a small window will have a lower air exchange rate than a small room with a large window. Large rooms, high ceilings, excessive glass, etc., will cause departures from these estimates.

Crack Method : The crack method calculates the flow produced by the pressure difference acting on each leakage path or building component. This method gives the higher accuracy, but, the major limitation is in estimating the appropriate pressure differences under appropriate design conditions of temperature and wind.

Infiltration causes both type of loads namely sensible load ($\dot{Q}_{is}$) and latent load ($\dot{Q}_{il}$). The loads due to infiltration are given by following expressions :

$$\dot{Q}_{is} = 1.232 \times L \times \Delta T \quad , (W) \tag{2.56}$$

and,

$$\dot{Q}_{il} = 3012 \times L \times \Delta\omega \quad , (W) \tag{2.57}$$

where, ΔT is difference between inside and outside air temperatures ^oC. $\Delta\omega$ is difference between inside and outside air specific humidities $kgw./kg\ d.a.$. L is infiltration air in litre per second.

$$L = \frac{1000 \times V_{room} \times N_{ach}}{3600 \times 2} \tag{2.58}$$

where, V_{room} is volume of the room and N_{ach} is the number of air changes per hour.

2.5 Ventilation Load

Ventilation air is mandatory to ensure fresh air supply in the residential buildings. The amount of ventilation needed has been debated for over a century,

and the different types of rationale developed have led to radically different ventilation standards. The current rationale [4] for the minimum outside air requirement is 2.5 L/s per person based on CO_2 concentration.

Ventilation, also, causes both types of load namely sensible load ($\dot{Q}_{vs}$) and latent load ($\dot{Q}_{vl}$). The load due to ventilation are given by following expressions :

$$\dot{Q}_{vs} = no \times 1.232 \times L \times \Delta T \quad ,(W) \tag{2.59}$$

and,

$$\dot{Q}_{vl} = no \times 3012 \times L \times \Delta\omega \quad ,(W) \tag{2.60}$$

where, ΔT is difference between outside and inside temperatures, ^oC. $\Delta\omega$ is difference between inside and outside air specific humidities, $kgw./kg\ d.a.$. L is ventilation air in litre per second per person and no is number of people in the conditioned space.

2.6 Occupancy Load

Human beings liberate heat and moisture which cause the sensible and latent heat load to the air conditioning system, respectively. The latent heat, thus, liberated can be considered as an instantaneous cooling load, but the sensible heat gain is not converted directly to cooling load. The instantaneous sensible load is the product of the sensible heat loss from the people and the cooling load factor (CLF). This CLF [5] is function of the time people spend in the conditioned space and the time elapsed since first entering. The sensible load ($\dot{Q}_{os}$) and latent load ($\dot{Q}_{ol}$) can be expressed as :

$$\dot{Q}_{os} = no \times Sensible\ heat\ loss \times CLF \quad ,(W) \tag{2.61}$$

and,

$$\dot{Q}_{ol} = no \times Latent\ heat\ loss \quad ,(W) \tag{2.62}$$

where, no is number of people in the conditioned space.

2.7 Lighting Load

An accurate estimate of the cooling load imposed by the lighting is not straight forward. Because the rate of heat gain to the air caused by lights can be quite different from the power supplied to the light points or lighting fixtures.

The time lag effect should be taken into account in calculating the cooling load, since the actual load is lower than the instantaneous heat gain, and peak load may be significantly affected.

The lighting load ($\dot{Q}_{ls}$) is expressed as :

$$\dot{Q}_{ls} = Light\ Wattage \times Special\ Allowance\ Factor \times CLF \qquad (2.63)$$

where, cooling load factor (CLF) [5] is a function of time of use , type of arrangement, room furnishing, etc. Special allowance factor is introduced for fluorescent fixtures and fixtures requiring more energy than their rated Wattage.

2.8 Power Equipment Load

Power equipment load ($\dot{Q}_{ps}$) is calculated from

$$\dot{Q}_{ps} = \frac{kW\ Rating \times Load\ Factor \times 1000 \times CLF}{\%Motor\ Efficiency/100}, (W) \qquad (2.64)$$

where, *load factor* is merely the fraction rated load delivered under the conditions of the cooling load estimate.

Cooling load factor (CLF) [5] is the function of time (Hours after equipments are on) and total operational time.

2.9 Appliances Load

In estimating cooling load, heat gain from heat producing appliances , e.g. kitchen appliances, computers, etc., must be taken into account. Some appli-

ances produce only sensible load while others generate sensible ($\dot{Q}_{as}$) as well as latent load ($\dot{Q}_{al}$).

$$\dot{Q}_{as} = Sensible\ Heat\ Rate \times CLF \tag{2.65}$$

and,

$$\dot{Q}_{al} = Latent\ Heat\ Rate \tag{2.66}$$

Here, it can be noticed that sensible load is present even after putting off the appliance while latent load is present only for that hours in which appliance is on.

2.10 Total Cooling Load

The solar load ($\dot{Q}_{ss}$) is the summation of

$$\dot{Q}_{ss} = \dot{Q}_{structural} + \dot{Q}_{glass} + \dot{Q}_{door} \tag{2.67}$$

Total cooling load is sum of all the sensible loads and latent loads.

$$\dot{Q}_{sensible} = \dot{Q}_{ss} + \dot{Q}_{is} + \dot{Q}_{vs} + \dot{Q}_{os} + \dot{Q}_{ls} + \dot{Q}_{ps} + \dot{Q}_{as} \tag{2.68}$$

and,

$$\dot{Q}_{latent} = \dot{Q}_{il} + \dot{Q}_{vl} + \dot{Q}_{al} + \dot{Q}_{oL} \tag{2.69}$$

Therefore,

$$\dot{Q}_{total} = \dot{Q}_{sensible} + \dot{Q}_{latent} \tag{2.70}$$

Chapter 3

Roof Surface Evaporation

Air cooling by water evaporation occurs in nature near waterfalls, flowing air streams over lakes and oceans, under summer showers and even upon wetted skin. The evaporative process simply removes sensible heat (i.e., cooling by decreasing the surface temperature) and replaces it with latent heat (i.e., increasing the moisture content of air)[14,15].

3.1 Theory

Evaporation is described as an adiabatic process, meaning that the total amount of heat in the thermal system remains constant. As water evaporates, the sensible heat content of the system falls, while the latent heat content increases by an equal amount. In other words, dry-bulb temperature of air falls, but its moisture content rises. The limit of temperature reduction is up to the wet-bulb temperature of the air at the beginning of the process. Surface evaporation can cool up to the wet-bulb temperature. The process stops when the relative humidity of air approaches 100 %. A simple measure of the potential for evaporative cooling at any given air condition is the wet-bulb depression, defined as the difference between the dry-bulb and wet-bulb temperatures. It provides an upper limit of the achievable temperature drop by

direct evaporation. Evaporative cooling may be :

- *Direct Cooling Process*
- *Indirect Cooling Process*

3.1.1 Direct Evaporative Cooling

In the direct evaporative cooling the water is evaporated directly by the air stream that flows into the conditioned space. In direct evaporative cooling equipment, water is supplied through a float valve to a small reservoir from where it is caused to flow down through fibrous pads. A fan draws large volumes of outdoor air through pads, where it is cooled by evaporation, and then supplied to the building. This cool and more humid air absorbs sensible heat from the building.

A perfect evaporative cooling process is illustrated in Fig. 3.1 . The outdoor conditions are represented by A on the psychrometric chart. The indoor conditions can be represented by C. Outdoor air is sent through the evaporative system and exit at saturation condition, point B. This idealized cooling process occurs along a line of constant wet-bulb temperature and stop at 100% RH. The humid, cool air at B is then mixed with internal air at C. The final indoor air condition will lie somewhere along the line B to C, its exact location depends on the relative amount of the two air volumes.

The temperature of air delivered by an evaporative cooler may be estimated by the following equation:

$$T_{supply} = T_{db} - (WB_{depression} \times \eta_{se}) \tag{3.1}$$

where,
η_{se}=saturation efficiency
$WB_{depression}$=$(T_{db}$ - $T_{wb})_{outside}$
T_{db}=outdoor dry-bulb temperature
T_{wb}=outdoor wet-bulb temperature

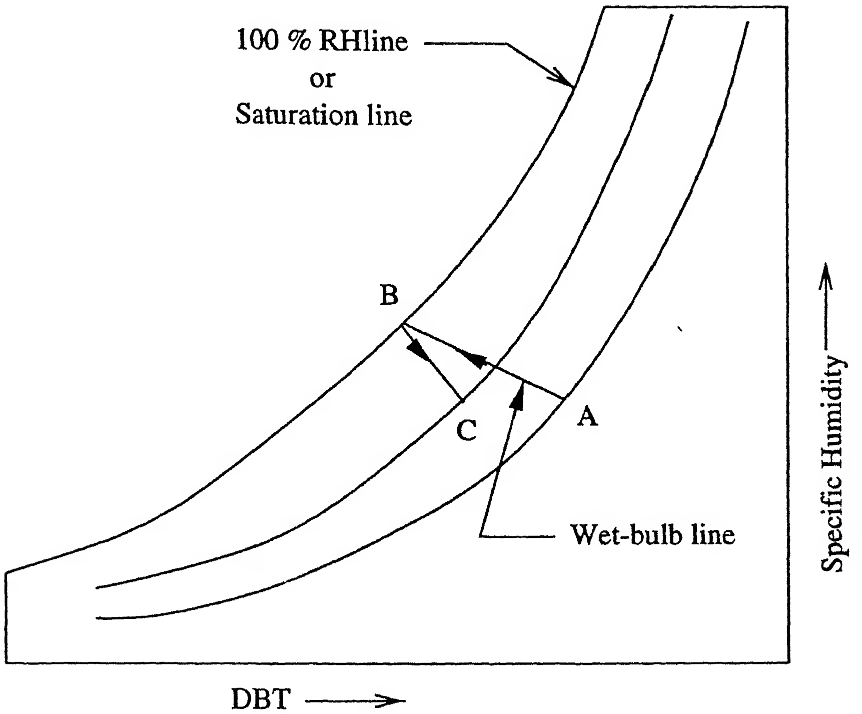

Figure 3.1: Direct Evaporative Cooling Process.

Direct evaporative cooling equipments humidify the air supplied to a building, rendering the relative humidity of indoors to be always higher than that of outdoors. The successful application of direct evaporative coolers depends on the existence of outdoor humidity levels well below human comfort conditions. This type of condition is specially prevalent in Northern India in summer.

3.1.2 Indirect Evaporative Cooling

In the indirect evaporative cooling water does not evaporate in the air suppled into conditioned space. It attempts to make use of evaporative cooling process without increasing the amount of moisture in the supplied air. Indirect evaporative cooling equipments use a heat exchanger to avoid the direct contact between water to be evaporated and the supply air. A direct evaporative process cools air that flows across one side of the heat exchanger, removing heat, and is then exhausted to atmosphere. The air to be supplied to the build-

ing flows across the other side of the heat exchanger and is cooled without receiving any moisture (Fig. 3.2).

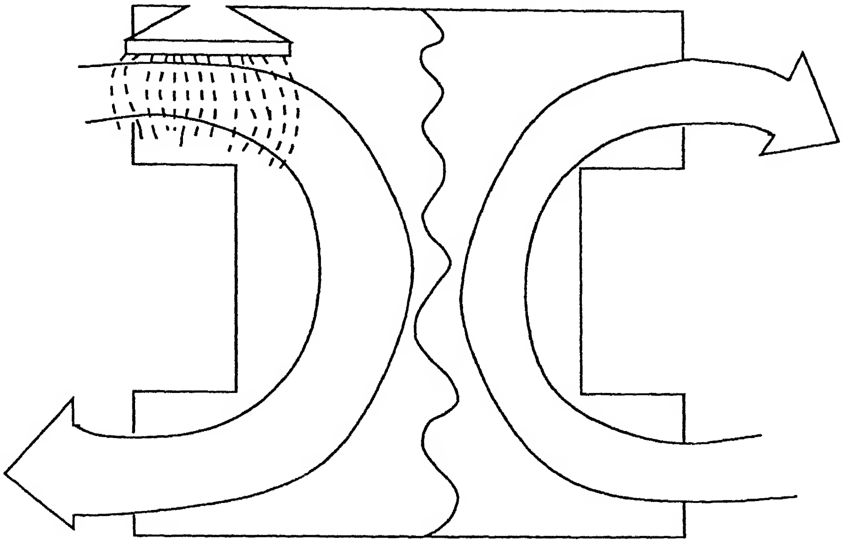

Figure 3.2: Indirect Evaporative Cooler Schematic.

The quality of water is immaterial in this case, as it never mixes with the supply air. Hence, even impure and waste water can be used for this purpose.

An indirect evaporative cooling process is illustrated in fig. 3.3 . The outdoor air at point A is cooled without the addition or removal of moisture along a horizontal line to point B. (The actual output conditions depend on the saturation efficiency of the direct evaporation process and the effectiveness of the heat exchanger.). Both the dry-bulb and the wet-bulb temperatures of the air are reduced; the moisture content of the air remains unchanged. But the dew-point temperature remains constant.

3.2 Roof Surface Evaporation

Roof surface evaporation [9], is a simple and passive process of cooling buildings, being effective, economically viable and practicable technology to provide the cooling of buildings using very small electric power. This process not only

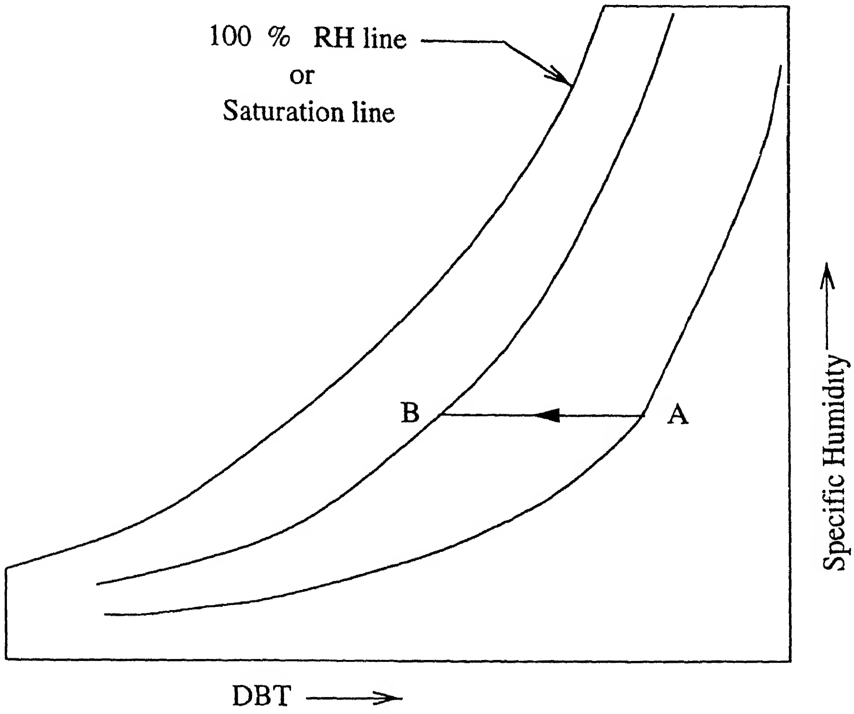

Figure 3.3: Indirect Evaporative Cooling Process.

eliminates major flow of heat to the building through its roof, but it may also extract heat from the building. Evidently, the surface evaporation brings the roof temperature close to the wet-bulb temperature. Hence it becomes a heat sink and so heat gain by the building through the walls, windows etc. is transferred to the roof, rendering reasonable comfort condition inside the building. This type of cooling is quite suitable for workshops, industrial buildings, theaters, etc.

Figure 3.4 shows a schematic diagram of the roof surface evaporation. It comprises a gunny bag layer over the roof, water sprinklers at proper intervals having pipe connections, a pump to maintain adequate pressure in the sprinklers and a water tank.

The evaporative cooling needs small quantity of water up to 9 liters per square metre of roof surface area per day under severe hot-dry weather conditions. Water has to be sprayed uniformly over a thin absorbing material lining provid-

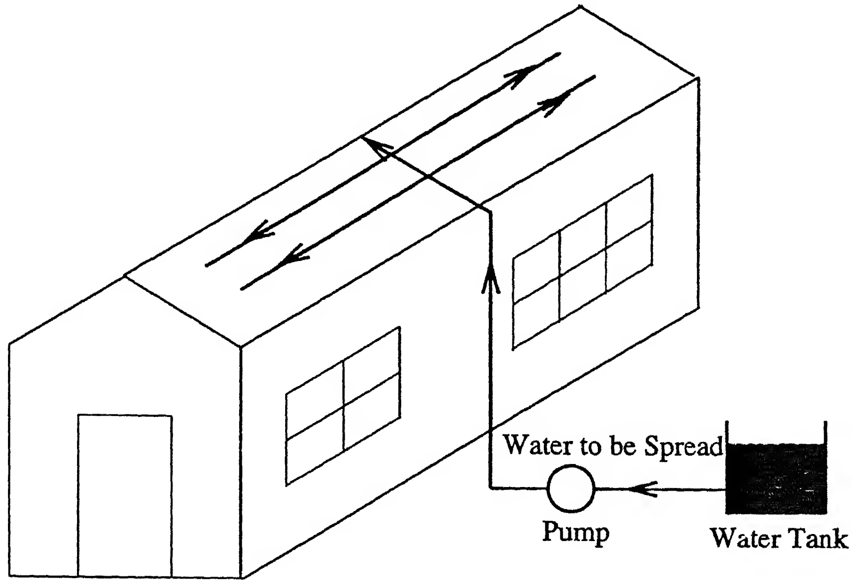

Figure 3.4: Schematic Arrangement for Roof Surface Evaporation.

ing wick action on the entire roof surface. It must absorb sufficient water and evaporate it quickly. The heat and water absorbing material lining is required to be maintained moist day and night during summer periods for continuous and quick evaporation.

The past manual spraying on empty cement bags for low cost housing schemes has been replaced by highly refined system to meet the requirement of sophisticated modern buildings. This has led to development of automatic spraying device and coirmating of the surface as heat absorptive material lining. Such arrangement is reliable and renders better performance. Quality of water do not make much difference for this cooling process except replacement of metallic sensing probes in automatic spraying system.

Recent development of hand-operated device and its successful implementation at various places, has made possible for easy adoption of this process by masses even in remote areas where both electricity and tap water may not be available.

The precautions, which have to be taken care of, particularly for water-proofing aspects at the design stage by changing the topography of the roofs more-

sloppy pitched or cylindrical dome to avoid stagnation of water on roofs.

The use of conventional heat insulating materials ,e.g., lime concrete, mud phaska, thermocole having so many practical problem can be straight way eliminated once this process is made integral part of buildings.

In the present work, mathematical model for *roof surface evaporation* is proposed for the residential buildings.

3.3 Heat Transfer through Roof with Surface Evaporation

In this case, also, the finite difference approach is used to calculate the inside surface temperature (t_{si}) and inside film coefficient (h_i) and hence, heat transfer through roof. The only difference arises in the way how equivalent temperature has been calculated.

Equivalent temperature for without surface evaporation:

$$t_e = T_{db} + \frac{\alpha_s \times I_t}{h_o} \tag{3.2}$$

Equivalent temperature for with surface evaporation:

$$t_e = T_{supply} + \frac{\alpha_s \times 0.0}{h_o} \tag{3.3}$$

where,

$$T_{supply} = T_{wb} + (1 - \eta_{se}) \times (T_{db} - T_{wb}) \tag{3.4}$$

η_{se} = saturation efficiency

Here, this analysis shows that roof surface evaporation will always fructify to provide low energy cooling even in humid climate as it eliminates the effect of solar radiation

Heat transfer through roof with surface evaporation ($(\dot{Q}_{roof})_{with\ evap.}$) is given as

$$(\dot{Q}_{roof})_{with\ evap} = h_i \times (t_{si} - t_i) \tag{3.5}$$

To calculate the total cooling load with surface evaporation ($(\dot{Q}_{total})_{with\ evap.}$), $\dot{Q}_{roof}$ is replaced by $(\dot{Q}_{roof})_{with\ evap}$.

Chapter 4

Computer Simulation for Cooling Load Calculation

A generalized computer program has been developed to predict the hourly cooling load for a residential building of any type of orientation. To facilitate the users, there are guidelines for the input data for getting

- hourly cooling load
- cooling load due to windows and doors
- cooling load due to infiltration and ventilation
- the miscllanious load comprising occupancy, lighting, appliances, power equipment, etc.

4.1 Inside Film Coefficients for Walls/Roof

For the inner surfaces of the enclosed space, the inside film coefficient is found to be the function of the inside air velocity and the temperature difference existing between the room air and inner surface of the walls/roof. The Newton

Raphson iterative procedure has been used to get inner surface temperature of the walls/roof and inside film coefficient.

The salient features are:

- Set intial condition, i.e.,

$$t_{si} = a\ large\ value, \quad (say, 1000.0) \tag{4.1}$$

- Calculate inside film coefficient $(h_i(j))$
- Find an objective function

$$f_n = B_{ii} \times t_{si}(j) + 2 \times t_{si} - B_{ii} \times t_i - 2 \times t_3(j) \tag{4.2}$$

where,

$$B_{ii} = \frac{h_i(j) \times \Delta x_3}{k_3} \tag{4.3}$$

$h_i(j)$ = inside film coefficient for walls/roof, taken from equations 2.21 and 2.22, respectively.
Δx_3 = thickness of the inner layer of walls/roof.
k_3 = thermal conductivity of the inner layer of walls/roof.
t_i = inside temperature.

- Calculate the derivative of objective function w.r.t. inside surface temperature

$$\frac{df_n}{dt_{si}} = B_{ii} + t_{si} \times \frac{dB_{ii}}{dt_{si}} + 2. - t_i \times \frac{dB_{ii}}{dt_{si}} \tag{4.4}$$

where the derivative of B_{ii} w.r.t. t_{si} is found from equation 4.3 as :

$$\frac{dB_{ii}}{dt_{si}} = \frac{\Delta x_3}{k_3} \cdot \frac{dh_i}{dt_{si}} \tag{4.5}$$

where the derivative of h_i w.r.t. t_{si} is found from equation 2.22 or 2.23, respectively for walls/roof and is expressed as :

$$\frac{dh_i}{dt_{si}} = 1.77 \times 0.25 \times (t_{si}(j) - t_i)^{(-0.75)} \quad \text{, (for Walls).} \tag{4.6}$$

$$\frac{dh_i}{dt_{si}} = 1.31 \times 0.25 \times (t_{si}(j) - t_i)^{(-0.75)} \quad \text{, (for Roof).} \tag{4.7}$$

- Calculate

$$dummy = t_{si}(j) - \frac{f_n}{df_n/dt_{si}} \tag{4.8}$$

- Compare, if $|dummy - t_{si}(j)|$ is less than some small value (say, 0.001) then Newton Raphson iteration converges for a particular time, else more iterations are performed by setting t_{si} equal to 'dummy'.

4.2 Inside Film Coefficient for Window Glass

For the window glass, the inside film coefficient is found to be function of the inside air velocity and the temperature difference existing between the room air and window glass. In this case, also, Newton Raphson iterative procedure has been used.

The salient features are:

- Set intial condition, i.e.,

$$t_g = a\ large\ value, \quad (say, 1000.0) \tag{4.9}$$

- Calculate inside film coefficient $(h_i(j))$
- Find an objective function

$$f_n = t_g(j) - \frac{X}{Y} \tag{4.10}$$

where,

$$X = A_{sun}(j) \times \alpha_D \times I_D + A_{win} \times \alpha_d \times I_d + A_{win} \times h_i(j) \times t_i + A_{win} \times h_o \times t_o(j) \tag{4.11}$$

$$Y = A_{win} \times (h_i(j) + h_o) \tag{4.12}$$

where, h_o and $h_i(j)$ are taken from equations 2.20 and 2.22, respectively.

- Calculate the derivative of objective function w.r.t. glass temperature

$$\frac{df_n}{dt_g} = 1.0 - \frac{Y.dX/dt_g - X.dY/dt_g}{Y^2} \tag{4.13}$$

where,

$$\frac{dX}{dt_g} = A_{win} \times t_i \times \frac{dh_i}{dt_g} \tag{4.14}$$

$$\frac{dY}{dt_g} = A_{win} \times \frac{dh_i}{dt_g} \tag{4.15}$$

where the derivative of h_i w.r.t. t_g is found from equation 2.21 and is expressed as :

$$\frac{dh_i}{dt_g} = 1.77 \times 0.25 \times (t_g(j) - t_i)^{(-0.75)} \tag{4.16}$$

- Calculate

$$dummy = t_g(j) - \frac{f_n}{df_n/dt_g} \tag{4.17}$$

- Compare, if $|dummy - t_g(j)|$ is less than some small value (say, 0.001) then Newton Raphson iteration converges for a particular time, else more iterations are performed by setting t_g equal to 'dummy'.

4.3 Computer Program for Cooling Load

The computer program is prepared based on the detailed expression put forth in chapter 2 and 3. For this, first, the building geometry and orientation are specified. Inside design condition is specified on the basis of the particular application. Other input values are taken as per the problem.

A flow chart is given in figure 4.1 and its details are outlined below :

- Calculation are carried out for first hour starting from midnight for occupancy load ($\dot{Q}_{os}$, $\dot{Q}_{ol}$), power equipment load ($\dot{Q}_{ps}$), lighting load ($\dot{Q}_{ls}$), appliances load (Q_{as}, $\dot{Q}_{al}$).

- Calculate outside temperature (t_O).
- Calculations are carried out for infiltration load ($\dot{Q}_{is}$, $\dot{Q}_{il}$) and ventilation load ($\dot{Q}_{vs}$, $\dot{Q}_{vl}$).
- Calculate solar radiation intencity (I_t).
- Calculations are carried out for solar load without evaporation ($\dot{Q}_{ss}$) and solar load with evaporation ($(\dot{Q}_{ss})_{evap.}$).
- Finally, $\dot{Q}_{total}$ and $(\dot{Q}_{total})_{evap.}$ are calculated.
- These calculations are repeated for each hour of the day till 24 hours.

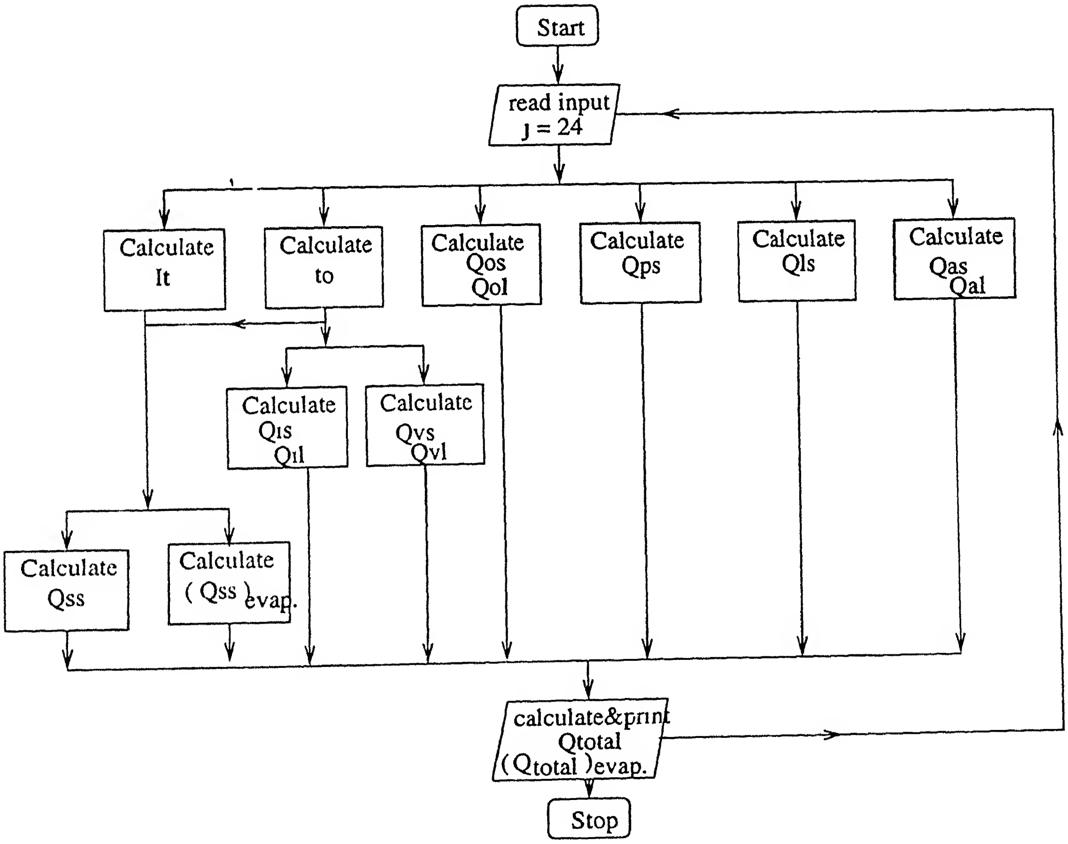

Figure 4.1: Flow Chart for Cooling Load Calculation.

Chapter 5

Results and Discussion

This chapter deals with results obtained by calculation of cooling loads on the basis of the theoretical and actual temperature variation of outside air. For this a generalised computer programme is made based on mathematical formulation discussed in chapter 2 and 3. The salient features of the present calculations are highlighted for Kanpur, though the method can be applied for any place by incorporating the data of the same. It also exhibits as to how the roof surface temperature gets reduced and the effect of radiation nullified. The energy conservation due to surface evaporation is shown through graphs. It is also discussed the effect of various sets of inside conditions in lieu of further reduction in cooling load without affecting the human comfort inside the room.

5.1 Input Data

The following results have been obtained for a particular room of Visitors Hostel in IITK, but the room has been isolated from the rest of the building for the worst case analysis and therefore to get the true effect of surface evaporation in percentage reduction of cooling load. These input values are given in Appendix 'E'.

5.2 Graphical Representation of Results

5.2.1 Theoretical and Actual Temperature Variation

The actual hourly temperature variation as well as the suggested hourly variation in temperature based on maximum and minimum temperature variation of the day have been shown in figures 2.1 to 2.4. The theoretical temperature generally matches very well with the actual values except deviating here and there with a few degree celsius. The deviation between them are such that over prediction at one time gets nullified by under prediction at other time. Hence, the present approach simplifies calculation procedure significantly.

5.2.2 Variation of Wet-Bulb Temperature

Figure 5.1 shows the variation in wet-bulb temperature for the month of April, May, June and July for Kanpur city. There is not much variation in WBT in a particular day. The maximum WBT occurs at about 3 P.M. except in July, in which the maxima occurs at about 10 A.M.. This much of variation may be different for different places.

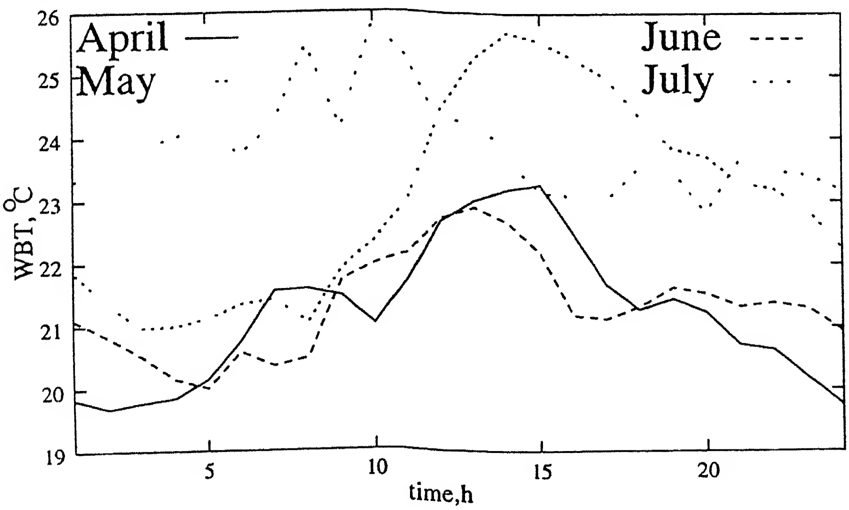

Figure 5 1: Hourly variation of wet-bulb temperature.

5.2.3 Variation of Relative Humidity

Figure 5.2 shows variation in relative humidity with time for the month of April, May, June and July for Kanpur city. The relative humidity for these months is minimum at about 3 P.M. when dry-bulb temperature is maximum.

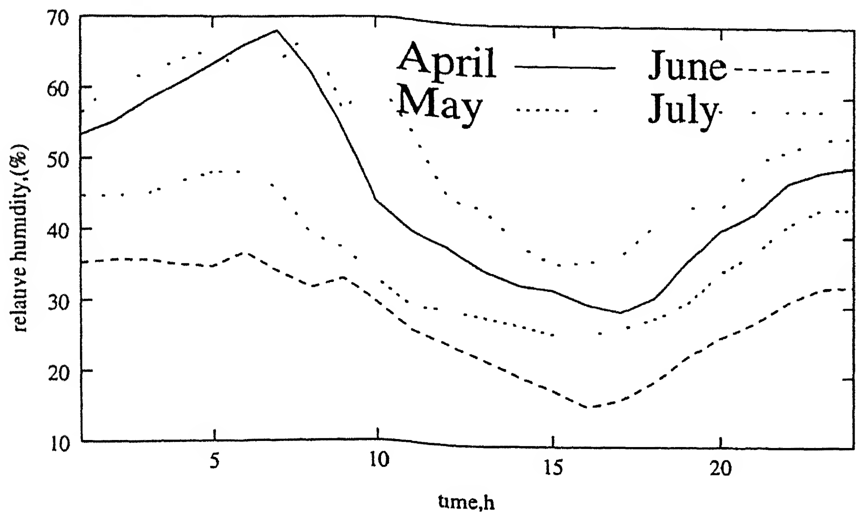

Figure 5.2: Hourly variation of relative humidity.

5.2.4 Sol-air Temperature Variation For Different Walls and Roof

The ambient temperature and sol-air temperatures for the different walls and the roof are presented in figure 5.3. The typical result is presented for the month of May. Evidently, it is seen that the sol-air temperature for a horizontal surface (roof) is highest among walls and roof. The same corresponds to the maximum incident solar radiation. As the sol-air temperature forms the driving potential for heat transfer, the same is found to be the dominant parameter governing the rate of heat transfer. Further, the heat transfer through the east and west facing walls is prominently higher than that of south facing walls. The heat transfer through the north wall is least among them. Thus

the area of eastern and western walls should be less than that of northern and southern walls.

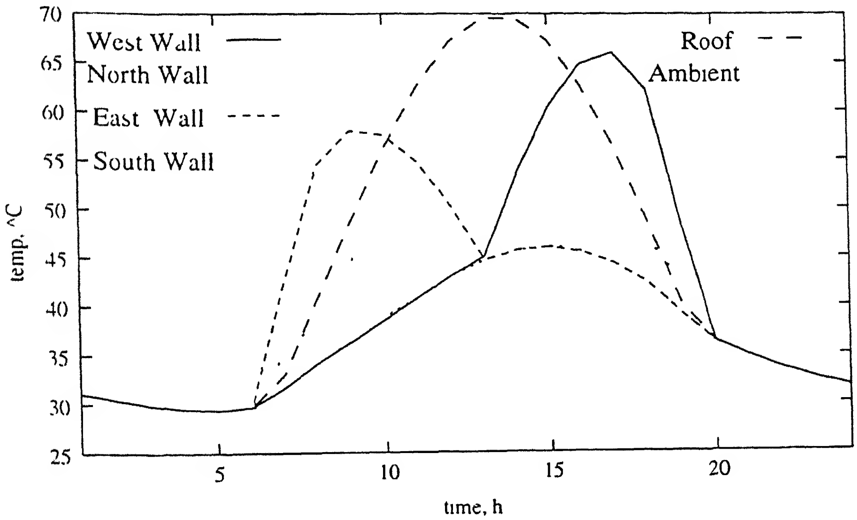

Figure 5.3: Variation of ambient temperature and sol-air temperature for walls and roof for the month of May.

5.2.5 Sol-air Temperature Variation of Roof alongwith WBT

Figure 5.4 shows sol-air temperature variation of roof and wet-bulb temperature for the month of May. If surface evaporation is applied on the roof, the roof surface temperature tends to wet-bulb temperature. As a result, there is substantial reduction in roof surface temperature, rendering reduced cooling load by applying surface evaporation. Interestingly, the surface evaporation is most suitable for applications where quite stable and uniform temperatures are essentially required.

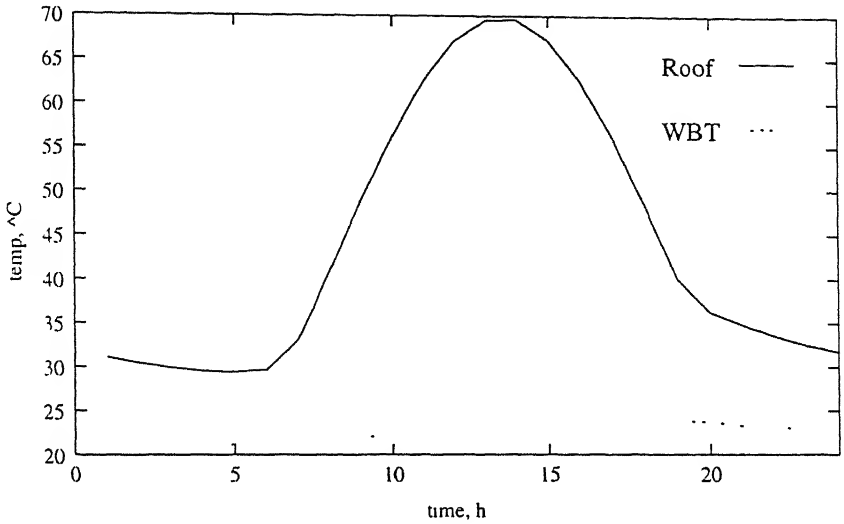

Figure 5.4: Variation of sol-air temperature of roof and wet-bulb temperature for the month of May.

5.2.6 Cooling Load with Actual and Theoretical Outside Air Temperature

In the current program coding, outside hourly ambient temperature has been calculated based on maximum and minimum outside temperature of particular day. This eliminates the need to collect the hourly temperature data. This has been justified by the fact that the cooling load with actual and theoretical temperature has been found very close to each other. This model is under predicting the cooling load by 3.14% for a typical month of May. Figure 5.5 shows the same comparison

5.2.7 Solar Load with and without Roof Surface Evaporation

Figures 5 6 to 5.9 show the hourly solar radiation with and without surface evaporation for the months of April, May, June and July respectively. These figures clearly show the exact amount of reduction in solar load by the use of roof surface evaporation

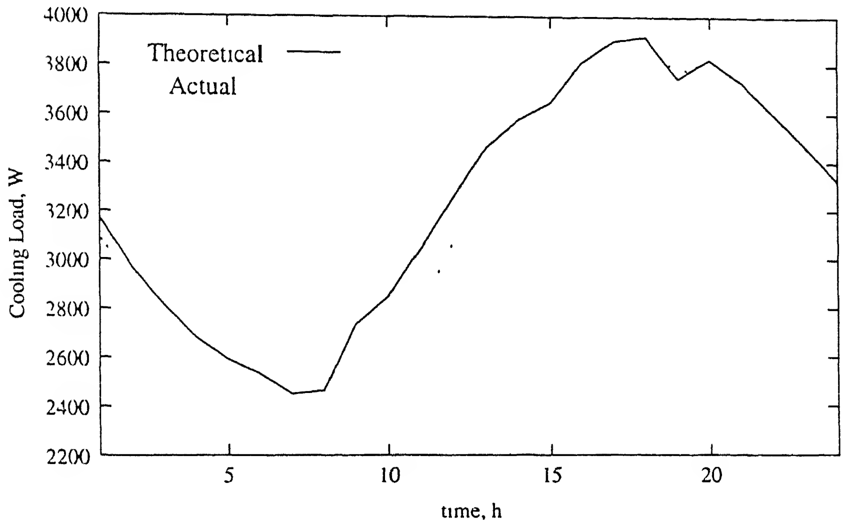

Figure 5.5. Variation of cooling load with actual and theoretical outside air temperature for the month of May.

5.2.8 Cooling Load with and without Roof Surface Evaporation

Figures 5 10 to 5.13 show the hourly variation of cooling loads with and without surface evaporation respectively for the months of April, May, June and July.

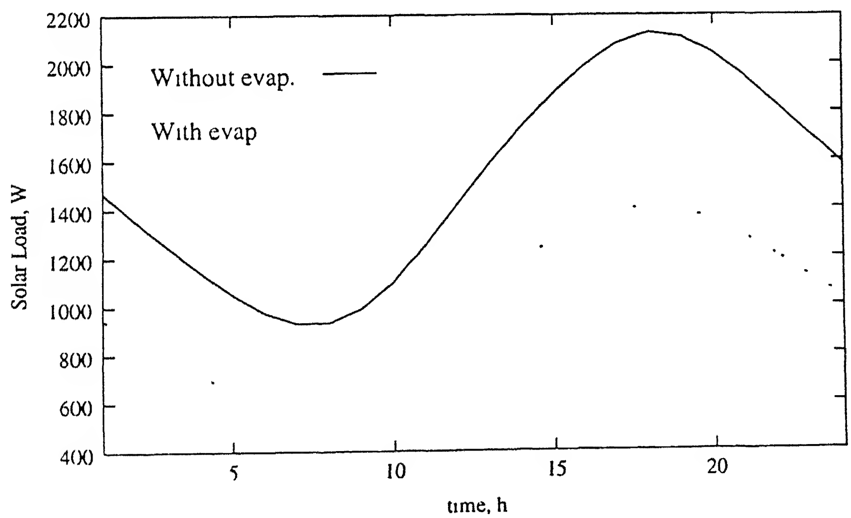

Figure 5 6: Variation of solar load with and without roof surface evaporation for the month of April

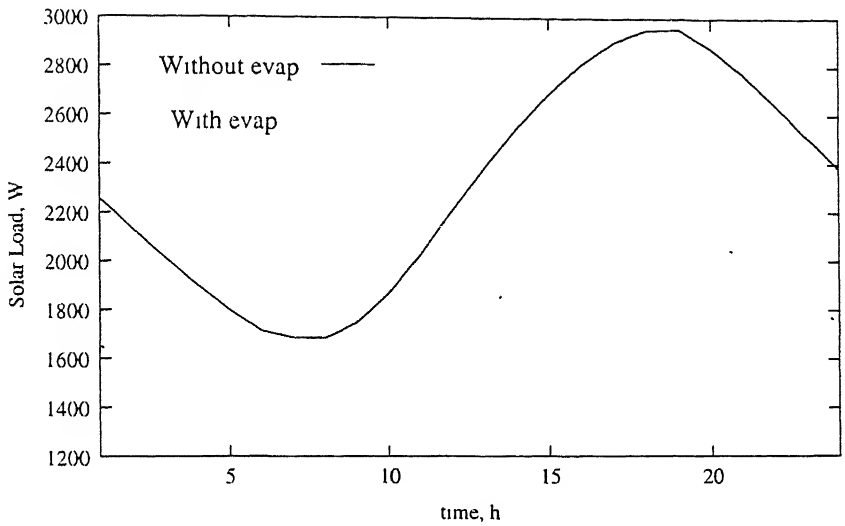

Figure 5.7 Variation of solar load with and without roof surface evaporation for the month of May.

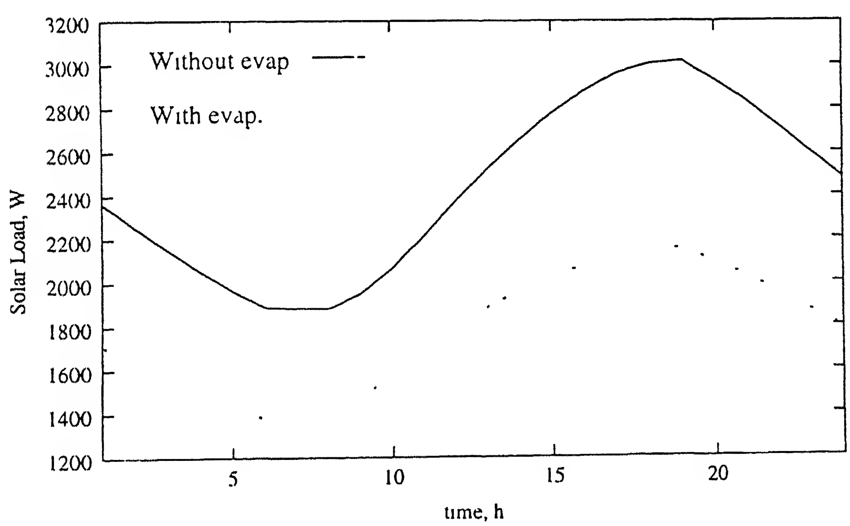

Figure 5.8: Variation of solar load with and without roof surface evaporation for the month of June.

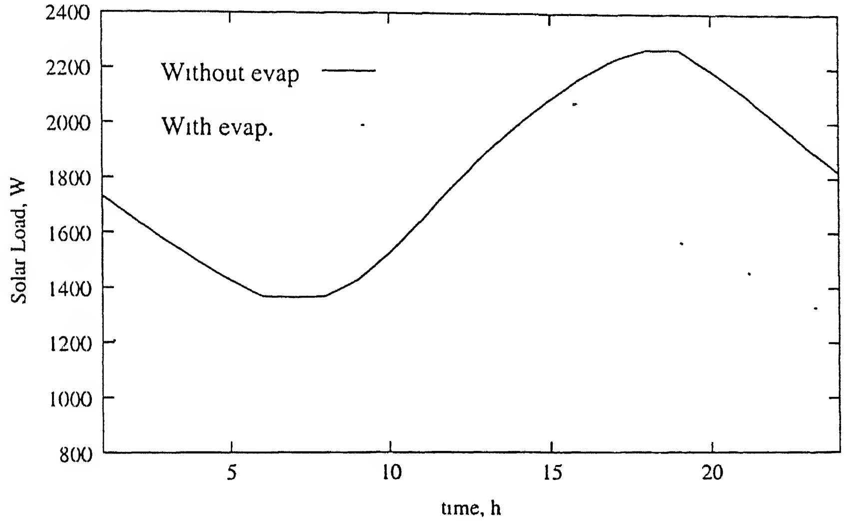

Figure 5 9 Variation of solar load with and without roof surface evaporation for the month of July

It is clear that hourly variation of cooling load is almost constant with surface evaporation. This is supported by figure 5.1 where the hourly variation in wet-bulb temperature is found to be very small as compared to the ambient dry-bulb temperature.

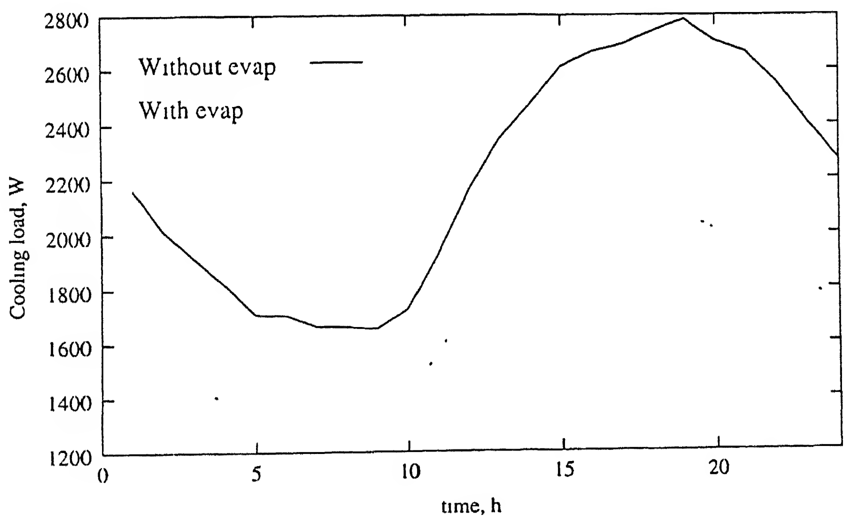

Figure 5.10. Variation of cooling load with and without roof surface evaporation for the month of April.

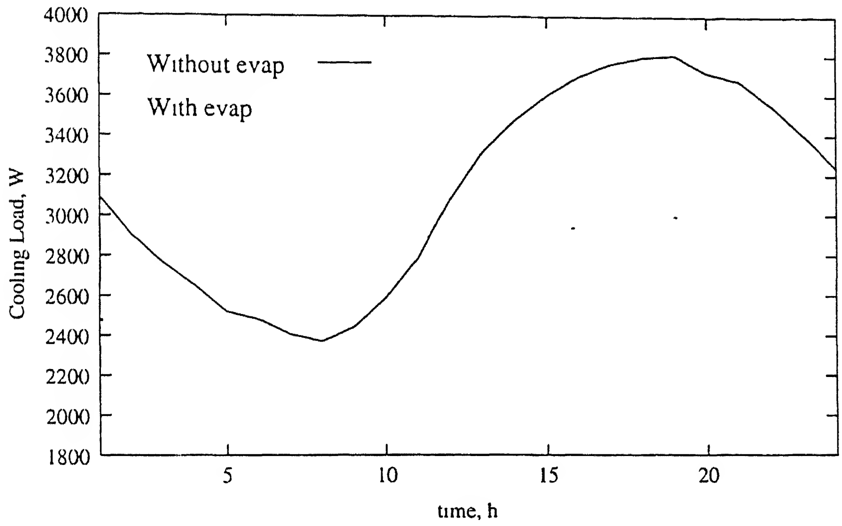

Figure 5.11. Variation of cooling load with and without roof surface evaporation for the month of May.

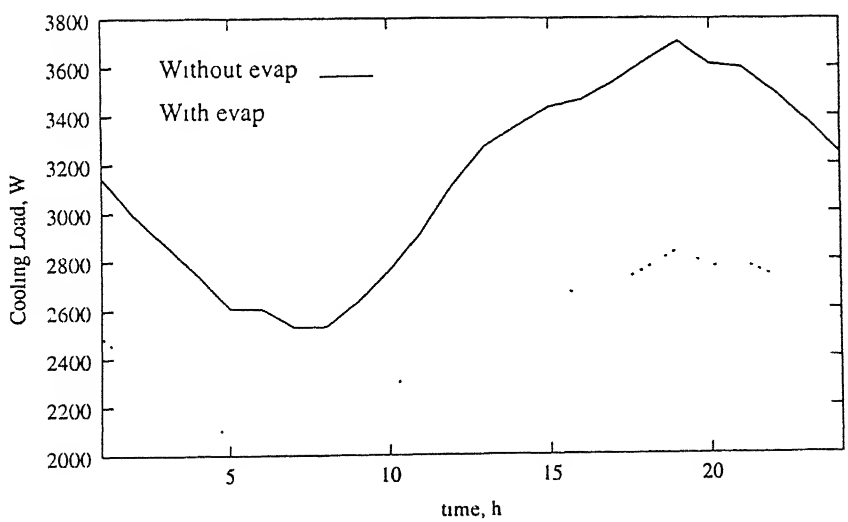

Figure 5 12· Variation of cooling load with and without roof surface evaporation for the month of June.

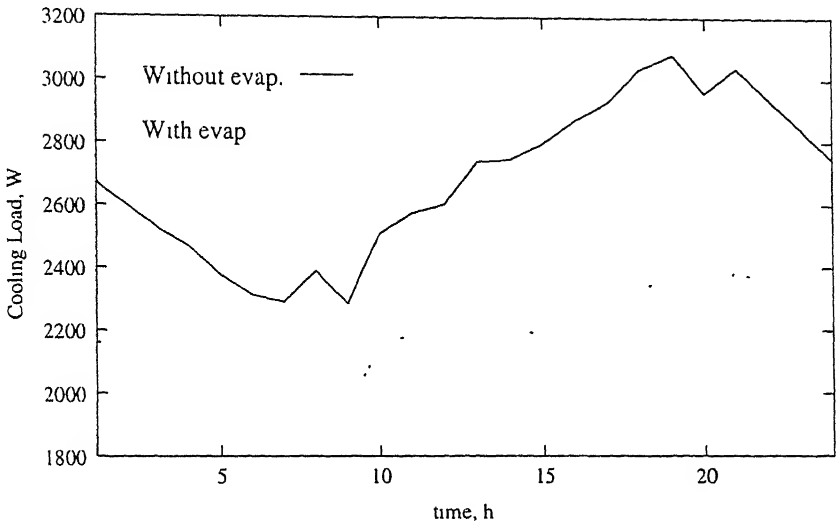

Figure 5.13: Variation of cooling load with and without roof surface evaporation for the month of July

5.2.9 Cooling Load for Various Humidifying Efficiencies

Figure 5.14 shows the hourly cooling load for the typical month of May with varying humidifying efficiencies It is noticeable that the effect of humidifying efficiency on cooling load is not much as roof surface evaporation nullify the solar radiation which is the main cause of cooling load, while humidifying efficiency changes the temperature differential across the roof which has less say in the cooling load.

5.2.10 Cooling Load for Various Inside Conditions

Cooling loads have been calculated for various inside conditions as suggested by different researchers keeping the comfort requirement in the mind. The table 5.1 gives the various sets of inside conditions, which have been considered in the present analysis.

Figure 5.15 shows the hourly variation in cooling load for the month of May with these different sets of initial conditions. It is noticeable that a significant

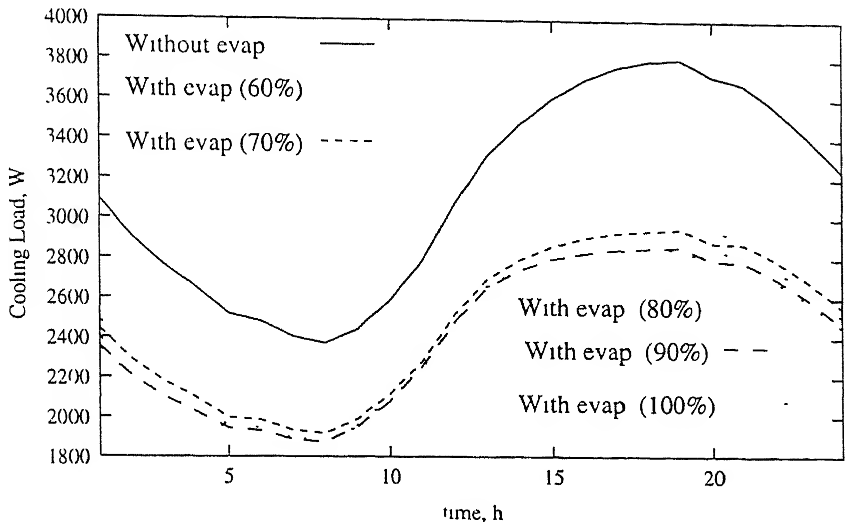

Figure 5.14. Variation of cooling load for different humidifying efficiencies for the month of May.

Table 5.1: Various sets of inside conditions.

Parameter	Set 1	Set 2	Set3
$t_i (^oC)$	25	27	29
$RH_i (\%)$	50	50	50
$V_i (m/s)$	0.13	0 50	1.20

reduction in cooling load is observed. Hence, in the era of energy conservation, this aspect should not escape attention while considering of a building construction.

5.3 Tabulated Results

Table 5.2 shows maximum calculated cooling loads for different months with and without roof surface evaporation. The roof surface evaporation decreases the structural load, while the rest of the components of cooling load remains unchanged.

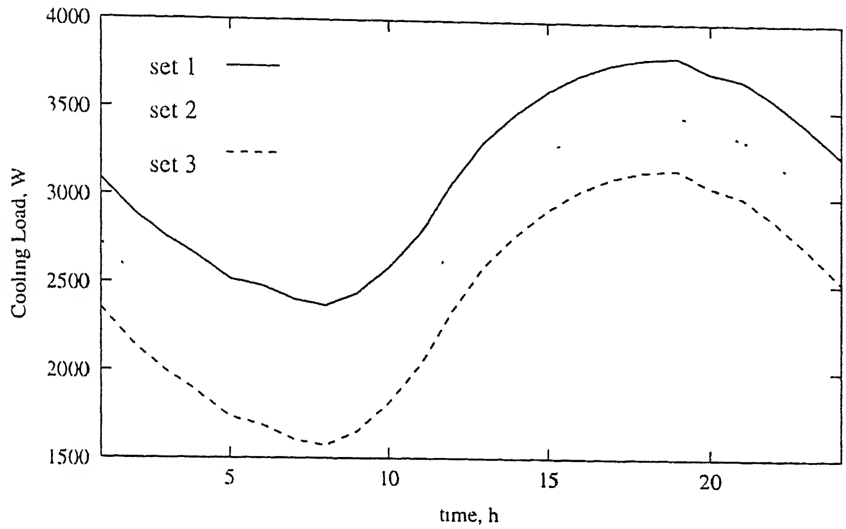

Figure 5 15: Variation of cooling load with various inside conditions for the month of May.

Table 5.2: Structural, occupancy, ventilation & infiltration, miscllaneous and total load.

Month	Structural Load		Occupancy Load	Ventilation & Infiltration Load	Misc. Load	Total Load	
	Without Evap.	With Evap.				Without Evap.	With Evap.
April	2108 96	1394 07	353.80	232 24	87.50	2782 50	2067.61
May	2957 29	2147.28	353 80	407.96	87 50	3806.55	2996.55
June	3015.80	2154.99	353.80	237.83	87.50	3704.12	2843.30
July	2270.09	1568.10	353.80	370.80	87.50	3082.19	2380.20

The total cooling loads with and without surface evaporation, and percentage saving in peak cooling load are shown in table 5.3 for a typical month of May. Roof surface evaporation renders about 21.3% saving in cooling load.

Table 5.3: Cooling load with and without surface evaporation and saving.

Total cooling load without surface evaporation	3806 55 W
Total cooling load with surface evaporation	2996.55 W
Saving in cooling load	810.00 W
Percentage of saving	21.3 %

Table 5.4 shows the maximum cooling load requirement for different months with different sets of inside conditions. With higher inside temperature alongwith enhanced air velocity, significant reduction in cooling load is observed.

Table 5.4: Cooling load with different sets of inside conditions.

Month	Total Cooling Load (W)		
	Set 1	Set 2	Set 3
April	2782.50	2437.31	2093.09
May	3806 55	3480.04	3161.99
June	3704.12	3377.00	3057.79
July	3082.19	2736.83	2391.77

Chapter 6

Conclusions and Suggestions

6.1 Conclusions

The following are the salient features of the present work :

1. The total cooling load requirement has been predicted on the basis of hourly temperature variation based on T_{max} and T_{min} for a given day. The same is compared with actual cooling load based on actual hourly temperature variation. The versatility of this is by the fact that T_{max} and T_{min} values are reported in several data books [5,16,17], but the hourly variation is available only at selected meteorological centers.

2. The inside film coefficient is predicted accurately by considering the dependence on the inside surface temperature and air motion. Newton-Raphson method has been used in this regard.

3. The cooling load for the summer months is found to be maximum in the month of June. This is in tune with the recommended month for practice engineers by the standard texts.

4. The higher indoor dry-bulb temperature is recommended with enhanced air velocity for the reduction in cooling load.

Chapter 6

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3. The cooling load for the summer months is found to be maximum in the month of June. This is in tune with the recommended month for practice engineers by the standard texts.

4. The higher indoor dry-bulb temperature is recommended with enhanced air velocity for the reduction in cooling load.

5. Surface evaporation is applied to the roof. It reduces the structural load considerably, with some calculated results render reduced cooling load to the tune of 21 %.

6. The roof is recommended to have 10 – 15^o tilt like hut, in order to dispense with seepage caused by water logging due surface evaporation system over the roof. This should be kept in mind while constructing new buildings.

7. The use of surface evaporation renders quite stable and uniform temperature irrespective of large range of variation in dry-bulb temperature of outside environment.

6.2 Scope for Future Work

1. This computer program to calculate the cooling load is applicable to the buildings located in the Northern Hemisphere. This program can be suitably modified for the buildings located in the Southern Hemisphere by changing the subroutine which calculate the solar radiation intensity.

2. The problem can be extended for the relative humidity and outside wind velocity data similar to one used for the ambient temperature. Thus, one dose not need to collect the meteorological data for various places.

3. Economic analysis can be done for the surface evaporation to find out initial investment and running cost, though it can be intuitively concluded that same will be far less than the cost of refrigeration system and its maintenance, cost for energy, etc.

4. An extensive experimental work should be carried out alongwith a planned publicity for the adoption the present system in order to solve the shortage of energy problem.

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Appendix A

Percentage of the Daily Range

Table A.1: Percentage of the Daily Range

time,h	%	time,h	%	time,h	%	time,h	%
1	87	7	93	13	11	19	34
2	92	8	84	14	3	20	47
3	96	9	71	15	0	21	58
4	99	10	56	16	3	22	68
5	100	11	39	17	10	23	76
6	98	12	23	18	21	24	82

Appendix B

Temperature and Relative Humidity of Outside Air

Table B.1: Hourly Outside Temperature (^oC) for Kanpur

time,h	Apr	May	June	July	Aug	Sep	Oct
1	28.2	31.5	33.2	30.9	31.6	31.3	22 0
2	27.0	30.7	33.0	30.4	31.5	31.0	21.6
3	26.7	30.0	32.3	29.9	31.4	30.4	21.5
4	25.2	29.4	31.7	29.8	30.8	30.7	18.8
5	25.2	29.8	31.4	29.4	30.8	30.9	19.8
6	25.3	29.8	31.6	29.4	30.9	30.4	20.2
7	25.2	30.4	31.8	29.5	31.4	30.5	20.3
8	27.4	32.2	32.5	30.2	31.7	32.4	22.5
9	30.6	36.2	35.3	31.8	33.9	33.7	25.9
10	31.6	37.5	36.4	33.0	34.3	34.3	26 9
11	33.1	39.5	37.9	32.8	34.8	35.1	28.3
12	34.6	40.4	38 8	33.7	31.9	36.0	29.6
13	35.9	41.5	40.2	34.3	35.9	36 1	29.9
14	36.1	42.0	40.8	34.2	36.0	36.4	32 3
15	36.3	41.8	41.1	34.8	36.0	36 6	32.6
16	36 6	42 4	41 1	34.9	36.6	35.9	32.9
17	36.1	41.8	41.3	35.2	34.8	35.3	30.2
18	35.2	40.2	40.3	34.5	34.4	34.3	27.7
19	33.4	36.3	38.2	33.5	33.6	33.9	26.1
20	32.3	36.9	36 8	32.8	33.1	33.2	25.3
21	31.2	34.8	35.5	32.1	32.9	32.7	24.1
22	30.5	33.5	35.0	31.3	32.6	32.4	23.9
23	29.6	32.7	34.0	31.3	32.1	32.2	22.9
24	28 9	32.2	33 7	31.1	31.7	31.9	22.6

Table B.2: Hourly Outside Relative Humidity (%) for Kanpur

time,h	Apr	May	June	July	Aug	Sep	Oct
1	53.3	44.7	35.3	56.3	88.7	82.3	78.0
2	55 3	44.7	35.7	60.7	89.3	86.3	83.3
3	58.3	45.0	35 7	62 0	91.7	85.7	84.3
4	60.7	46.7	35.0	64.0	92.0	85.7	84.0
5	63.3	48.0	34 7	65 3	90.3	84 0	87.7
6	66.0	48.0	36.7	62.0	88.7	80 0	88.0
7	68.0	45.7	34.0	63.3	74.7	76.3	84.7
8	62 3	39.3	31 7	68 0	81.0	70.0	71.3
9	54.0	37.3	33.0	56.7	78.7	60.0	56.7
10	44.0	33 0	29.7	61 7	77.0	54.7	50.0
11	40 0	29 3	26.0	54.3	69.7	54.0	46.3
12	38.0	29.3	24.3	45.0	66.0	51.7	40.3
13	35.0	28.3	22 3	43.7	63.3	51.0	37.0
14	33.0	27 3	20 0	38.7	60 0	50.7	34.7
15	32.3	26.0	18.0	36 0	60.7	49 0	30.7
16	30.3	26.0	15.7	36 3	61.7	53 3	35.7
17	29.3	17 0	16.7	37.3	62 7	58.7	41.0
18	31.3	28 3	19.3	41.7	68.7	69.3	50.0
19	36.7	30 7	23.0	44.3	71.0	72.0	53.3
20	41.0	35 0	25.7	44.0	77.7	75.0	65 3
21	43.3	37 7	27.7	50.7	84 0	74.0	71 3
22	47 7	41 7	30.7	52.3	85.7	80.0	72.0
23	49.3	44 0	32.7	54.0	85 3	81.7	73 3
24	50.0	44.0	33.0	54.3	85.3	80 3	75 3

Appendix C

Constants for Solar Radiation Calculation

Table C.1: Constants for Solar Radiation Calculation

Month	Equation of Time min.	A W/m^2	B (Dimensionless Ratios)	C
Jan	-11.2	1230	0.142	0.058
Feb	-13.9	1214	0.144	0.060
Mar	-7.5	1185	0.156	0.071
Apr	1.1	1135	0.180	0.097
May	3.3	1103	0.196	0.121
Jun	-1.4	1088	0.205	0.134
July	-6.2	1085	0.207	0.136
Aug	-2.4	1107	0.201	0.122
Sep	7.5	1151	0.177	0.092
Oct	15.4	1192	0.160	0.073
Nov	13.8	1220	0.149	0.063
Dec	1.6	1233	0.142	0.057

A = Apparent Solar irradiation at zero air mass , (W/m^2)

B = Atmospheric extinction coefficient

C = Diffuse radiation factor

Appendix D

Properties of Ordinary Glass

Table D.1: Properties of Ordinary Glass for Direct Radiation

Angle of Incidence	0^o	10^o	20^o	30^o	40^o	50^o	60^o	70^o	80^o	90^o
Transmissivity, τ_D	0.87	0.87	0.87	0.865	0.86	0.84	0.79	0.67	0.42	0.00
Absoptivity, α_D	0.05	0.05	0.05	0.055	0.06	0.06	0.06	0.06	0.06	0.00

For diffuse radiation, regardless of the angle of incidence

$\tau_d = 0.79$

$\alpha_d = 0.06$

Appendix E

Input Data for Residential Building

Room Size	4.65$\times$ 3.80$\times$ 3.15 m^3.
Thickness of Wall	0.275 m.
Thickness of Roof	0.150 m.
Window size (on North Wall)	1.20$\times$ 0.74 m^2.
Door size (one, on North & South Wall)	2.1$\times$ 0.90 m^2.
Inside Relative Humidity	50 %
No. of Air Changes per hour	1.5
No. of Occupants	4
No. of Fluorescent Tubes	2
Rating of Fluorescent Tubes	40 W
No. of Lamp	1
Rating of Lamp	60 W
No. of Fan	1
Rating of Fan	75 W